INTRODUCTION
TO
NAVAL ARCHITECTURE

J. P. COMSTOCK

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Preface

THIS book is substantially the course in theoretical naval architecture given to the hull drawing apprentices at the shipyard of the Newport News Shipbuilding and Dry Dock Company. Its purpose is to acquaint the student with the fundamentals of theoretical naval architecture, and to explain how they are interrelated and how they are progressively applied in the design of a ship.

The book is not intended to displace the standard reference books on the subject; on the contrary, it is designed to lead the students to such books. It is written, not as a reference book for a practicing naval architect, but as a study book for a high school graduate. It first applies each subject to a definite "example" ship, with problems applying generally to this same ship, and then works through a "problem" ship somewhat different from the "example" ship.

A practical working knowledge will of course require much further study and experience; this is only an introduction.

Acknowledgment is due to the management of the Newport News Shipbuilding and Dry Dock Company for permission to publish this material.

John P. Comstock

Introduction

BEFORE finishing this book the student will follow through the basic design of a ship; that is, he will choose the main characteristics of the ship so as to satisfy the owner's requirements. But before he can do this, he must know something of the various calculations which are made, several times over, during a design; first very approximately, then, as the design takes definite shape, more and more accurately. The first twelve chapters are devoted to these subjects, using as a basis an example ship.

This example ship, particulars of which are given in Chapter I, is intended to represent the data on ships which have been built which every naval architect's office must have for reference.

Textbooks are referred to at the beginning of each chapter, reference to which will add greatly to the student's understanding. This book may, however, be completely worked through without any of these except:

"The American Bureau of Shipping Rules," necessary for Chapter V.
Taylor's "Speed and Power of Ships," for Chapters IX and X.
"Load Line Regulations," for Chapter XI.

A knowledge of elementary trigonometry is assumed.

Problems are given throughout the book. Working these problems will both test the student's understanding of the subjects and tend to fix them in his mind. The serious student is advised to work every problem.

In the last four chapters, a problem ship, somewhat different from the example ship, is presented and the basic design worked through, to show the trial-and-error methods and the occasional back-tracking involved in such work.

Contents

PART I—SHIP CALCULATIONS

PART II—DESIGN

Part I

SHIP CALCULATIONS

Chapter I

DEFINITIONS, ABBREVIATIONS AND PARTICULARS OF EXAMPLE SHIP

Definitions

Note: In the following chapters, terms which have been defined below will be used without further definition, except where additional explanation is considered necessary.

After perpendicular. A line perpendicular to the base line through the after edge of the rudder post, or, in the case of a cruiser stern, through the centerline of the rudder stock.

Amidships. In the vicinity of the middle portion of a vessel. Also, a point exactly half way between the forward and after perpendicular.

Appendages. Such items as shafting, struts, bossings, bilge keels, rudder, etc., extraneous to the hull and generally immersed.

Area of sections. The area of any transverse cross section of the immersed part of a vessel below a specified waterline.

Base line. A straight line from which the ordinates of a curve are measured. Also, a horizontal plane through the lowest point of the molded form amidships.

Batten. A smooth limber stick used for drawing curved lines.

Battens (cargo). Planks usually about 2 inches thick, fitted on the inside of the framing in holds for the protection of cargo.

Beam, molded. The width over the widest portion of the molded form.

Bending moment. The bending effect of external forces on a structure.

Body plan. A drawing showing transverse sections

through a ship; either the stations used in displacement calculations, or sections taken at every frame.

Bonjean curves. A set of curves giving the area of each transverse section up to any height above the base line.

Bulkhead. A vertical structure dividing a ship into compartments.

Bulkhead deck. The deck to which the main transverse bulkheads which divide the ship into watertight compartments are carried.

Buoyancy. The supporting force exerted by a liquid upon a body wholly or partially immersed in it. It is equal to the weight of the displaced liquid.

Buoyancy, reserve. The additional buoyancy that would result if that part of a vessel's hull which is above the load waterline were immersed.

Buttocks. Lines showing the shape of longitudinal vertical planes through the molded form.

Camber. The transverse curvature of a deck, making the centerline higher than the sides.

Center of buoyancy (*CB*). The center of gravity of the displacement.

Center of buoyancy, longitudinal (*LCB*). The location longitudinally of the center of buoyancy.

Center of buoyancy, vertical (*VCB*.) The height of the center of buoyancy above the base line.

Center of flotation (*CF*.) . The center of gravity of the waterplane of the vessel.

Center of gravity (*G*). The point at which all of the individual items making up a weight, volume or area may be considered as concentrated without changing their combined moment about any axis.

Center of gravity, longitudinal (*LCG*). The location longitudinally of the center of gravity.

Center of gravity, vertical (*VCG*). The height of the center of gravity above the base line.

Classification. Certification by a classification society as to the character of construction and outfitting of a vessel.

Classification society. An institution that supervises the construction of vessels under established rules, tests material for hulls, machinery and boilers, proof tests anchors and anchor chains and issues certificates of classification.

Coefficient. A ratio between certain characteristics of a vessel which serves as a means of comparing that vessel with others.

Coefficient, Admiralty $= \frac{\Delta^{2/3} V^3}{ihp \text{ or } shp}$. A coefficient used in approximate power estimating. It is the cube root of the square of the displacement in tons times the cube of the speed in knots divided by the indicated or shaft horsepower.

Coefficient, block $= \frac{Volume}{L \times B \times d}$. The ratio of the immersed volume of a ship to the product of the length, breadth and draft.

Coefficient, displacement-length $= \frac{Displacement}{(L/100)^3}$. The ratio of a vessel's displacement, in tons of salt water, to the cube of its length in hundreds of feet.

Coefficient of fineness. The ratio of the area under a curve to the area of its circumscribed rectangle. Also, the ratio of the volume of a solid to the volume of a circumscribed rectilinear parallelopiped.

Coefficient, longitudinal. See prismatic coefficient.

Coefficient,* midship section $= \frac{Area}{B \times d}$. The ratio of the immersed area of the midship section to the product of breadth and draft.

Coefficient,* prismatic $= \frac{Volume}{L \times midship\ area}$. The ratio of the immersed volume of a ship to the product of its waterline length and immersed area of midship section.

Coefficient, waterplane. The ratio of the area of a waterplane to its circumscribing rectangle.

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* If the maximum section is not amidships, the word maximum should be substituted for midship throughout these definitions.

Compression. Stress tending to shorten a member.

Corresponding speeds. Speeds which bear the same relation to each other as that of the square roots of the lengths of the ships involved.

Couple. Two equal and opposite forces not in the same straight line. Their resultant force is zero, but they exert a moment.

Cross curves of stability. A series of curves of righting arm plotted to a base of displacement, each curve being drawn for a given degree of heel.

Curve of sectional areas. A curve, plotted from a straight base line representing the length of the ship, the ordinates of which represent the areas of the vessel's immersed cross sections. The area under this curve represents the volume of this displacement. The center of gravity of this area represents the longitudinal center of buoyancy and the ratio of its area to that of the circumscribing rectangle is the prismatic coefficient.

Depth, molded. The vertical distance from the top of the beam of the uppermost continuous deck at the side of the vessel amidship to the top of the keel (base line).

Displacement. The amount of water displaced by a floating vessel. It equals the weight of the vessel itself with whatever is on board.

Draft. The depth of a vessel below the waterline measured vertically to the lowest part of the hull.

Draft marks. Numbers placed at the bow and stern of a vessel to indicate the draft.

Draft, mean. The mean of the drafts at the bow and stern.

Drag. The designed excess of draft aft over that forward.

Elastic limit. The stress at which a material ceases to return to its original dimensions when the load is removed.

Equilibrium. A condition in which the sum of all external forces and the sum of all external moments acting on a body are both equal to zero.

Equilibrium, neutral. A state of equilibrium in which a body displaced from its original position by an external force

tends to maintain the displaced position after the force has ceased to act.

Equilibrium, stable. A state of equilibrium in which a body displaced from its original position by an external force tends to return to its original position after the force has ceased to act.

Equilibrium, unstable. A state of equilibrium in which a body displaced from its original position by an external force tends to depart farther from the original position after the force has ceased to act.

Equipment number or numeral. A number used by classification societies for determining the number and sizes of anchors, cables, hawsers, etc.

Factor of safety, nominal. The ratio of the ultimate strength of a part to the designed working stress.

Factor of safety, real. The ratio of the yield point of a part to the designed working stress.

Flare. The outward slope of the upper part of a vessel's sides.

Floodable length. That portion of the length of a vessel at any point which may be flooded without causing her to sink below the margin line.

Forward perpendicular. A line perpendicular to the base line through the forward side of the stem at the designed waterline.

Freeboard. The distance from the waterline to the top of the weather deck at the side.

Girder, ship or "hull girder." That portion of a ship's hull structure which is composed of continuous longitudinal members, which resist the forces which tend to produce hogging and sagging.

Hogging. A distortion of a vessel in which the bow and stern drop below their normal position relative to the midship portion of the vessel. Opposite of sagging.

Horsepower, effective. The horsepower required to overcome the resistance of the vessel through the water; that is, the tow-rope power.

Horsepower, indicated. The horsepower developed within a reciprocating engine.

Horsepower, frictional. The horsepower used in overcoming frictional resistance.

Horsepower, residuary. The horsepower used in overcoming residuary (wave-making, etc.) resistance.

Horsepower, shaft or brake. The power transmitted from the engine to the propeller through the shaft.

Houses. Structures above the upper or superstructure deck, not extending the full width of the ship.

Inclining experiment. An operation in which a slight list is obtained by means of a known weight for the purpose of determining the center of gravity of a vessel.

Integrator. An instrument for determining the area, moment and moment of inertia of a plane figure.

Intercostal. Not continuous.

Knot. A unit of speed of one nautical mile (6080 feet or one-sixtieth of a degree on the equator) per hour.

Length between perpendiculars. The length on the load line, from the fore side of the stem to the after side of the rudder post; where there is no rudder post, to the center of rudder stock.

Length, effective. The length used in form calculations and in speed and power calculations.

Length, overall. The total length from the foremost to the aftermost points of a vessel's hull.

Lines. The plans of a ship that show the form of its hull, consisting of four views: Body plan, waterlines, buttocks and diagonals.

List. A transverse inclination of a vessel.

Load line or load waterline. See "Waterline (load)."

Margin line. A line 3 inches below the top of the bulkhead deck at the side.

Metacenter. The point of intersection of a vertical line through the center of buoyancy of a ship in the position of equilibrium with a vertical line through the new center of buoyancy when the ship is slightly heeled.

Metacenter, longitudinal or transverse. The metacenter corresponding to longitudinal or transverse inclination.

Metacentric height. The vertical distance from the center of gravity to the metacenter. It is termed transverse or longitudinal according to whether transverse or longitudinal metacenter is used.

Metacentric radius. The distance from the center of buoyancy to the metacenter.

Midship section. The transverse section located at the mid-point between the forward and after perpendiculars. Also, a drawing showing the principal scantlings of a vessel.

Molded form. The form described by the ship's lines as laid down on the mold loft floor.

Moment about any axis. The product of a force, volume or area, and the distance from its center of gravity to that axis.

Moment of inertia about any axis. The sum of the products of each infinitesimal part of an area or weight times the square of the distance of that part from the axis.

Moment to change trim one inch. The moment, usually expressed in tons times feet, required to change the trim of a vessel by one inch.

Naval architect. One responsible for the design of ships.

Neutral axis. The line near the middle of a beam, which, when the beam is subjected to bending, is neither in tension nor compression. In straight beams the neutral axis passes through the center of gravity of the section.

Offset. A term used by draftsmen and loftsmen for an ordinate to ship lines.

Ordinate. The distance from a base line to a point on a curve.

Parallel middle body. That portion of a vessel's body throughout which the sections retain the same area and shape as the midship section.

Permeability. The percentage of a given space which can be occupied by water.

Permissible length. A legal limit of bulkhead spacing in

passenger vessels obtained by multiplying the floodable length by a factor of subdivision.

Planimeter. An instrument for measuring the area of a plane figure.

Propulsive coefficient. The ratio of *ehp* to *shp*.

Radius of gyration $= \sqrt{I/A}$. The distance from the center of gravity of a weight or area to the point where the entire weight or area could be concentrated without changing the moment of inertia. It is, therefore, equal to the square root of the quotient obtained by dividing the moment of inertia by the weight or area.

Racking. The tendency to change transverse shape, due to rolling, wind, etc.

Reserve buoyancy. See "Buoyancy, reserve."

Residuary resistance. The difference between the total resistance and the frictional resistance, consisting principally of wave-making resistance.

Resistance, air. Resistance due to the above-water portion of the vessel moving through air.

Resistance, eddy-making. Resistance due to the formation of eddies.

Resistance, frictional. Resistance due to the friction of the water upon the surface of the ship.

Resistance, wave-making. Resistance due to maintaining the wave system associated with the moving ship.

Righting lever or arm. The perpendicular distance between a vertical line through the center of gravity and one through the center of buoyancy.

Righting moment. The product of the displacement and length of the righting lever.

Sagging. A distortion of the vessel in which the midship portion sags below its normal position relative to the bow and stern. Opposite of hogging.

Scantlings. A term applied to the dimensions of the frames, girders, plating, etc., in a ship's structure.

Section-area. See "Area of sections."

Section modulus $= I/y$. The moment of inertia of a

section divided by the distance from the neutral axis to the most distant part of the section (that is, to the "extreme fiber").

Shear. Stress tending to cause two contiguous parts of a body to slide on each other.

Sheer. The vertical longitudinal curvature of a vessel's rail, deck, etc. Also, the amount by which the height of the weather deck at the forward or after perpendicular exceeds the height amidships.

Ship girder. See "Girder, ship."

Speed-length ratio. The ratio of the speed in knots to the square root of the effective length in feet.

Stability. The tendency of a vessel to return to her original position when inclined away from that position.

Stability, initial. Stability in the upright position.

Standard Series (Taylor's). A comprehensive series of models towed at Washington to determine values of residuary resistance.

Stress. Load or force per unit area.

Strain. Change in length per unit length due to stress.

Superstructure. A structure extending clear across the ship, immediately above the uppermost complete deck, as a forecastle, bridge or poop.

Tension. Stress tending to lengthen a member.

Tonnage, register. The interior volume of a ship (minus certain exemptions and deductions), expressed in tons of 100 cubic feet.

Tons per inch of immersion. The number of tons of additional weight required to immerse a vessel one additional inch of draft.

Trim. The difference between the forward and after drafts.

Trochoid. The curve traced by a point on a radius of a circle which is rolled along a straight line.

Ullage. The distance from the top of a tank to the top of the liquid in the tank.

Wake. The water near a vessel's stern which is given a forward velocity relative to undisturbed water by the passing of the vessel.

Waterline. The intersection of any waterplane with the molded form; often used for waterplane.

Waterline (load). The waterline at which a vessel floats when fully loaded.

Waterplane. Any plane parallel to the surface of the water and limited by its intersection with the vessel's hull.

Wetted surface. The area of the immersed surface of the hull.

Yield point. The stress at which a material begins to "flow" or undergo plastic deformation.

Abbreviations

A.......... Area of waterplane, or of a stressed section.
A.B.S....... American Bureau of Shipping.
AP........ After perpendicular.
B.......... Breadth or beam of ship, or
.... Center of buoyancy, ship erect.
B′......... Center of buoyancy, ship inclined.
BL......... Base line.
BM....... Distance from *B* to *M* (metacentric radius).
.... Bending moment.
BP......... Between perpendiculars.
CB......... Center of buoyancy.
CF......... Center of flotation (*CG* of waterplane).
CG......... Center of gravity.
CL......... Centerline.
D.......... Depth of ship.
d.......... Draft of ship, or
.... Diameter of propeller.
Δ (delta)... Displacement of ship in tons.
E.......... Modulus of elasticity.
ehp........ Effective horsepower.
ehp_f........ Frictional effective horsepower.
ehp_r........ Residuary effective horsepower.
f........... Coefficient of frictional resistance.
FP........ Forward perpendicular.
G.......... Center of gravity.

GM........Distance from *G* to *M* (metacentric height).
GZ.........Righting lever.
HB........Half-breadth.
hp.........Horsepower.
I...........Moment of inertia of stressed section, or
..........Moment of inertia of ship's waterplane.
i...........Moment of inertia of free surface of tank, etc.
ihp........Indicated horsepower.
K..........Keel, intersection of centerline and base line.
KB........Distance from *K* to *B*, or *VCB*.
KG........Distance from *K* to *G*, or *VCG*.
KM........Distance from *K* to *M*, or height of metacenter.
L..........Length of ship.
LBP.......Length between perpendiculars.
LOA.......Length overall.
LWL.......Length on waterline.
LCB.......Longitudinal center of buoyancy.
LCF.......Longitudinal center of flotation, or *LCG* of waterplane.
LCG.......Longitudinal center of gravity.
LI.........Longitudinal interval.
LM........Longitudinal metacenter.
l...........Length of beam or column.
l/r.........Slenderness ratio (see *l* and *r*).
M.........Metacenter, or
.........Bending moment.
M_t.........Transverse metacenter.
M_l.........Longitudinal metacenter.
M″.........Moment to trim one inch.
μ (mu).....Permeability.
NA........Neutral axis.
p..........Maximum stress in beam, column, etc., or
..........Pitch of propeller.
R..........Reaction, or
..........Resistance.
R_f.........Frictional resistance.
R_r.........Residuary resistance.
r..........Radius of gyration.

S (or SM)	Section modulus.
S	Wetted surface.
SM	Simpson's multipliers.
shp	Shaft horsepower.
Σ (sigma)	The sum of.
T	Thrust of propeller.
V	Volume of ship's displacement, or
	Vertical shear, or
	Ship's speed in knots.
v	Volume of tank, etc.
VCB	Vertical center of buoyancy.
VCG	Vertical center of gravity.
W	Weight.
WL	Waterline.
y	Distance from neutral axis to extreme fiber.

Description of Example Ship

Owner's Requirements

Deadweight (total)	8000 tons
Speed	12 knots
Draft	24 feet
No passengers, 45 crew	

Principal Dimensions, etc.

Length between perpendiculars	420 feet
Length, for displacement	415 feet
Beam	54 feet
Depth	30 feet 3 inches
Draft	24 feet
Displacement (block 0.76)	11,780 tons
Cargo capacity, general	400,000 cubic feet

Weights and Centers (for use in Chapter XI)

Steel:		
Main Hull	1980 tons	15.7 feet *VCG*
Erections	210	38
Houses and masts	160	43
Total steel	2350	19.3
Miscellaneous Hull:		
Hull fittings	116 tons	38 feet *VCG*
Carpenter work	120	20
Joiner work	48	38
Cement, tile and paint	105	10
Anchors, chains and lines	53	21
Stewards and deck outfit	16	30
Heat insulation	22	28
	480	25.1
Hull Engineering:		
Deck machinery	40 tons	34 feet *VCG*
Steering gear	8	27
Electrical plant	10	28
Ventilation	21	33
Plumbing	4	37
Pumping and drainage, heating, etc.	47	18
Miscellaneous	10	26
	140	27.1
Propelling machinery (3300 *ihp*)	610 tons	14.8 feet
Total light ship	3580	19.7 feet
Crew and stores	10	33
Fuel oil	1050	9
Feed water	210	3
Drinking and culinary water	100	12
Cargo	6750	18.5
Total deadweight (against 8000 tons required)	8120	16.8
Total displacement	11,700 tons	17.7 feet

Chapter II

LINES, FORM CALCULATIONS AND CURVES OF FORM

References

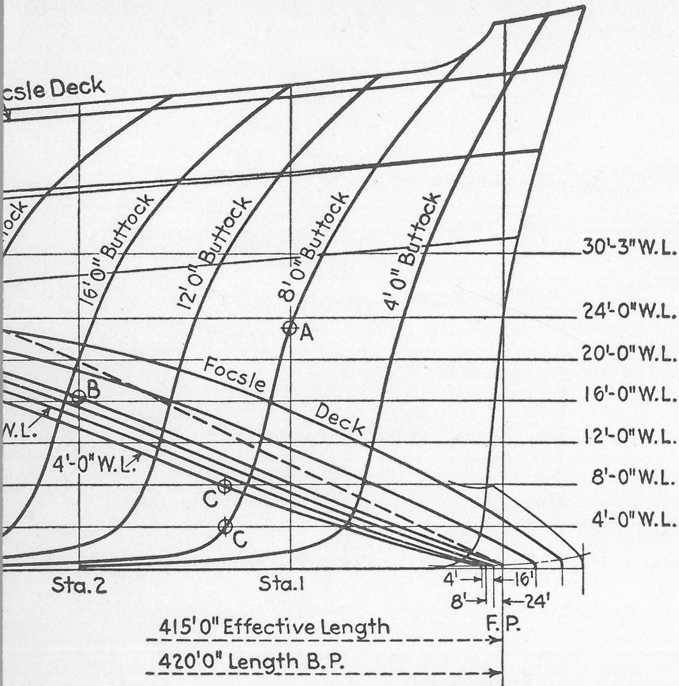
(*a*) "Principles of Naval Architecture," Volume 1, Chapter 1.
(*b*) Attwood's "Theoretical Naval Architecture."
(*c*) Peabody's "Naval Architecture."

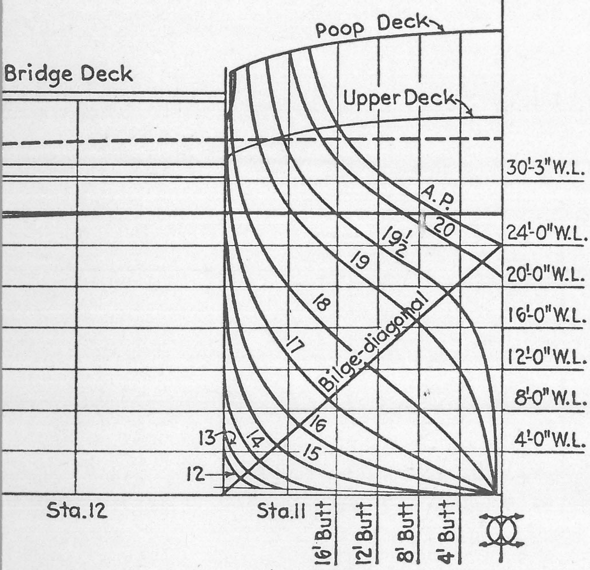
Lines

The "lines" of the example ship are shown on Fig. 1. In this figure, as is usual, the four views are drawn on a common base line for convenience. This base line represents the centerline of the ship when drawing the waterlines and the diagonal, and the base line of the ship when drawing the buttocks and the body plan. The length between perpendiculars is divided into 20 stations. The body plan is drawn with its centerline on station 10. One side only is drawn, as the ship is symmetrical about the centerline. The right-hand side of the body plan shows the fore-body stations; that is, those forward of amidships; and the left-hand side shows the after-body stations. This vessel has a parallel middle body extending from station 6 to station 12, so that in the body plan, stations 6 to 12 are alike.

These lines have been drawn to give a predetermined curve of sectional areas. In Chapter XV will be shown how this section-area curve is obtained. For the present, however, a brief description will be given showing how the lines drawing was made.

The required area of each station was obtained from the section-area curve. Each station was then lightly sketched in, free-hand, tested for area with the planimeter, and changed, still free-hand, until the area was correct. A sheer,





Afterbody Plan

Lines of Example Ship

Length B. P.	420' 0"
Length, Effective	415' 0"
Depth	30' 3"
Beam	54' 0"
Draft	24' 0"
Block Coefficient	
Sheer (Standard)	
Displacement Molded	
Midship Section Coefficient	
Prismatic Coefficient	

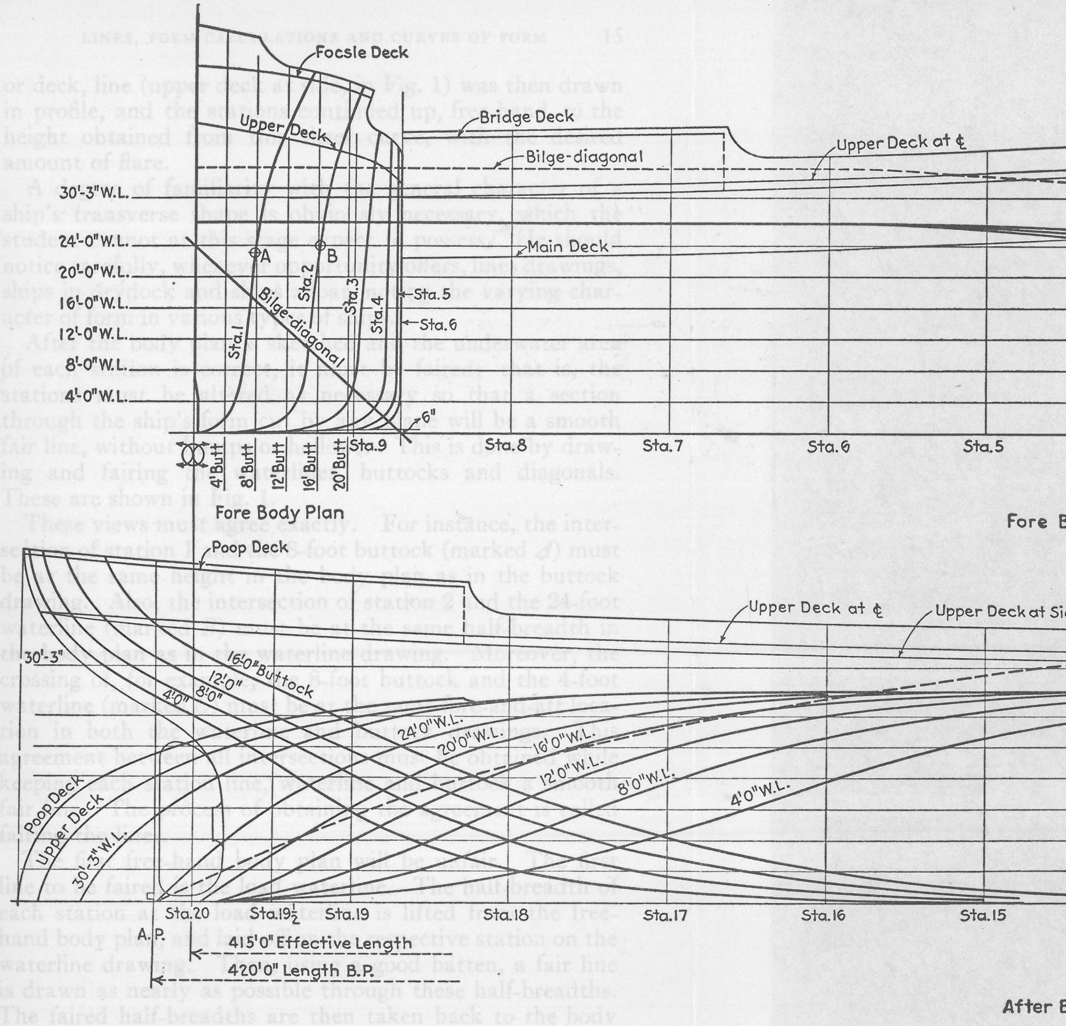
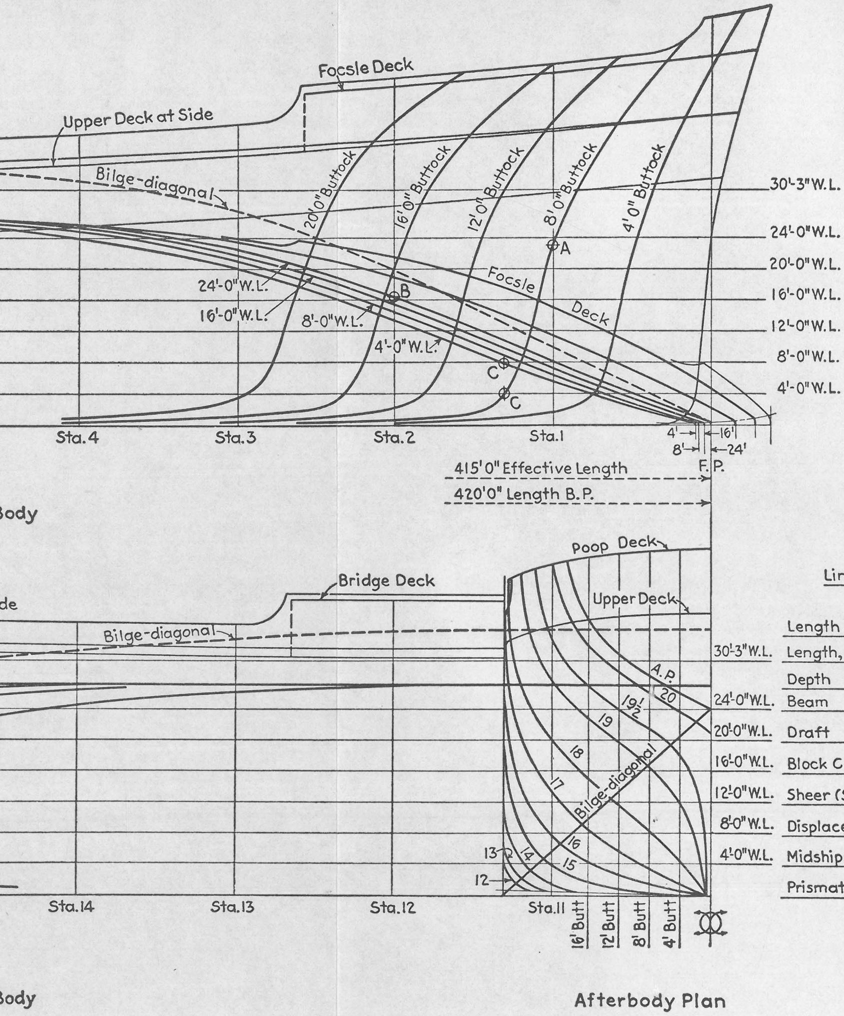

Focsle Deck
Upper Deck
Bridge Deck
Bilge-diagonal
Upper Deck at ¢
30'-3" W.L.
24'-0" W.L.
20'-0" W.L.
16'-0" W.L.
12'-0" W.L.
8'-0" W.L.
4'-0" W.L.
A
B
Main Deck
Sta.1
Sta.2
Sta.3
Sta.4
Sta.5
Sta.6
Bilge-diagonal
6"
4' Butt
8' Butt
12' Butt
16' Butt
20' Butt
Sta.9
Sta.8
Sta.7
Sta.6
Sta.5
Fore Body Plan
Fore B
Poop Deck
Upper Deck at ¢
Upper Deck at Si
30'-3"
16'0" Buttock
12'0"
8'0"
4'0"
24'0" W.L.
20'0" W.L.
16'0" W.L.
12'0" W.L.
8'0" W.L.
4'0" W.L.
Poop Deck
Upper Deck
30'3" W.L.
Sta.20
Sta.19½
Sta.19
Sta.18
Sta.17
Sta.16
Sta.15
A.P.
415'0" Effective Length
420'0" Length B.P.
After B

Fig. 1.—"Lines" o



Lines of Example Ship

Length B.P.	420' 0"
Length, Effective	415' 0"
Depth	30' 3"
Beam	54' 0"
Draft	24' 0"
Block Coefficient	
Sheer (Standard)	
Displacement Molded	
Midship Section Coefficient	
Prismatic Coefficient	

f the Example Ship

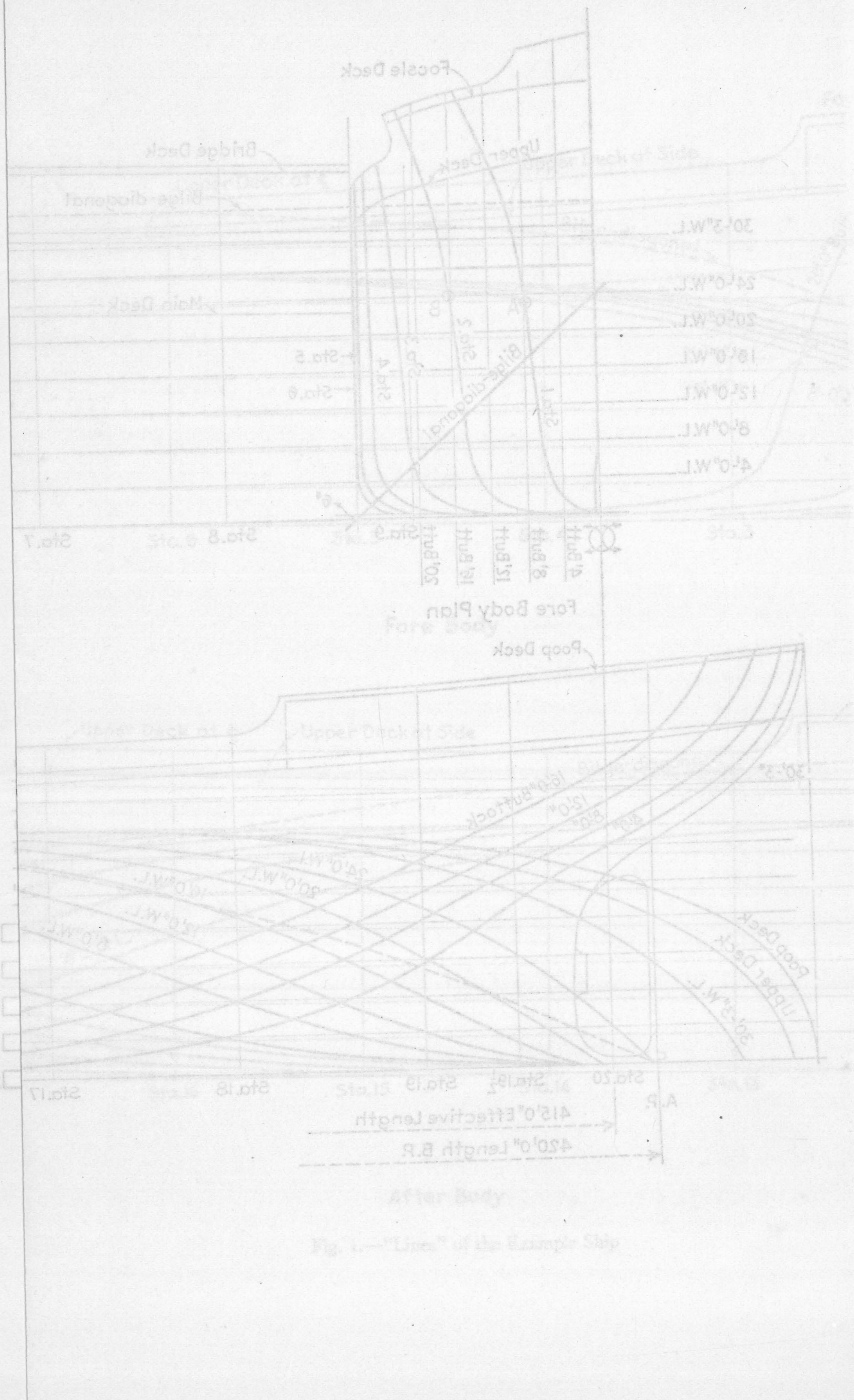


Fig. 1.—"Lines" of the Example Ship

or deck, line (upper deck at side, in Fig. 1) was then drawn in profile, and the stations continued up, free-hand, to the height obtained from this sheer curve, with the desired amount of flare.

A degree of familiarity with the general character of a ship's transverse shape is obviously necessary, which the student cannot at this stage expect to possess. He should notice carefully, whenever opportunity offers, lines drawings, ships in drydock and ships afloat, noting the varying character of form in various types of ships.

After the body plan is sketched and the underwater area of each station is correct, it must be faired; that is, the stations must be altered as necessary so that a section through the ship's form cut by any plane will be a smooth fair line, without humps or hollows. This is done by drawing and fairing the waterlines, buttocks and diagonals. These are shown in Fig. 1.

These views must agree exactly. For instance, the intersection of station 1 and the 8-foot buttock (marked *A*) must be at the same height in the body plan as in the buttock drawing. Also, the intersection of station 2 and the 24-foot waterline (marked *B*) must be at the same half-breadth in the body plan as in the waterline drawing. Moreover, the crossing of, for example, the 8-foot buttock and the 4-foot waterline (marked *C*) must be at the same fore-and-aft location in both the waterline and buttock drawings. This agreement between all intersections must be obtained while keeping each station line, waterline and buttock a smooth fair line. The process of obtaining the agreement is called fairing the lines.

The first free-hand body plan will be unfair. The first line to be faired is the load waterline. The half-breadth of each station at the load waterline is lifted from the free-hand body plan, and laid off on the respective station on the waterline drawing. Then, using a good batten, a fair line is drawn as nearly as possible through these half-breadths. The faired half-breadths are then taken back to the body plan and the stations changed to suit them. The next line

to be faired should be the bilge diagonal. Then the other waterlines are faired, until waterlines and station lines and the diagonal are in complete agreement.

In fairing the buttocks, which come next, a new complication is added. Not only must these pass through the heights lifted from the body plan, they must also cross the waterlines at points obtained from the waterline drawing, where the waterlines cross the buttocks (see point *C* on Fig. 1). The buttock is run through both sets of points as nearly as may be, and both waterline and station changed as necessary to suit the faired buttock.

The process is continued until complete fairness is attained.

Curves of Form. Curves of form are curves showing several useful characteristics of a vessel's underwater form. The calculations of the data from which these curves are plotted are called form calculations. Before discussing them some discussion of the plotting of curves in general may be helpful.

Any two quantities, one of which depends on the other, may be expressed by a curve. For instance, in Fig. 2 the displacement of a vessel, which depends on the draft, is plotted as a curve from which can be read the displacement corresponding to any draft. Similarly, a curve of power required to drive a vessel at any speed can be plotted on a base line of speed, as in Fig. 60.

The principal reason for plotting such curves is to get intermediate points without calculations. For example, in Fig. 2 the displacement would be calculated only at drafts several feet apart. Then, since we know from the smooth nature of the ship's form that the curve of displacement must be a smooth curve between the calculated points, the displacement for any draft can be read directly from the curve with no further calculation.

Another reason for plotting such curves is that the area under the curve may have especial meaning. For instance the area under a section-area curve, such as that shown in

Fig. 62, being the product of a length and ordinates which represent areas, represents a volume, in this case the volume of the displacement.

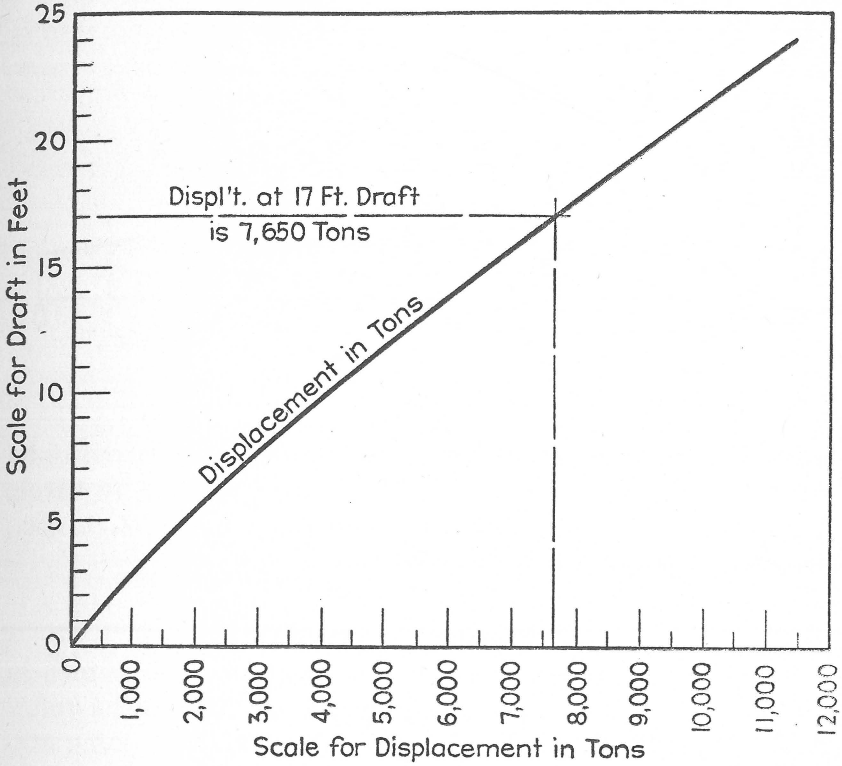


Fig. 2

Simpson's Rule. The areas of mathematical figures such as cylinders, cones, etc., can be calculated exactly. The areas of figures bounded by non-mathematical curves, such as ship's lines, a section-area curve, etc., require approximate methods, the commonest of which, in ship work, is Simpson's first rule, usually called just Simpson's rule. It can be proved [see Reference (*a*)] that if the base line of a smooth curve, such as is shown in Fig. 3, is divided into any even number of equal spaces, and the ordinates of the curve

measured at those divisions, the area bounded by the curve, its base line and its end ordinates can be found by multiplying the sum of the end ordinates, four times all the odd-

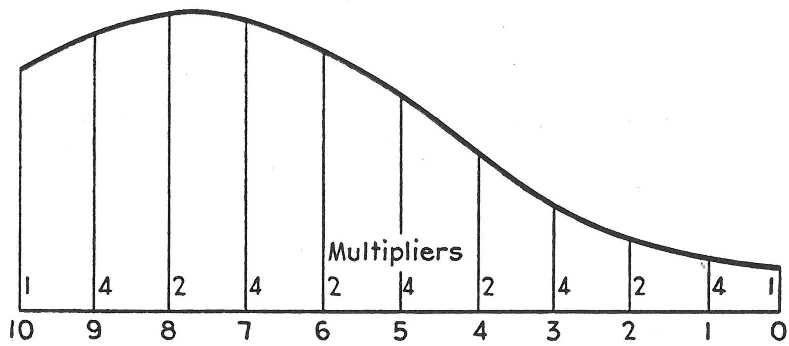


Fig. 3

numbered ordinates, and twice all the even-numbered ordinates, by the length of the spaces, and dividing by three. This is Simpson's rule, and the multipliers, 1, 4, 2, 4, etc., are called Simpson's multipliers.

Form Calculations

These are arranged as on the calculation sheet shown in Fig. 4. The columns are numbered for convenient reference. Twenty stations are used, which are the same stations used on the section-area curve and the body plan in Fig. 1.

Displacement (Columns 2 and 4 in Fig. 4). To obtain the areas of the stations, a planimeter is used. This is an instrument with a tracing point and a recording wheel, so built that, if the point is run around any closed figure, the reading of the wheel is proportional to the area of the figure. A planimeter is shown in Reference (*a*), page 22, and proof of its operation is given in Reference (*c*), page 11. The area of the figure is obtained by multiplying the planimeter reading by a constant which depends on the instrument and on

ple Ship) To 24'-0" W.L.

$\frac{\text{um}_6 \times 2 \times \text{L.I.}}{9}$ = Trans. Mom. of Inertia

$\frac{\text{om. of Inertia}}{\text{Cu. Ft. Displ.}}$ = Trans. B.M.

TRANS. B. M.		
Column No. (11)	(3)	(12)
B. of W.^{3}	S.M.	Products
Sum$_6$ =		

x $\frac{\text{Sum}_6 \times 2 \times \text{L.I.}}{9}$ = x $\frac{2}{9}$ x

SUMMARY		
Molded Displacement S.W.		
Shell & Appendg. Displ. (6%)		#
Total S.W. Displacement		⊕
Block Coeff.		⊕
Max. Section Coeff.		⊕
Prismatic Coeff.		⊕
Water Plane Coeff.		⊕
Tons per Inch S.W.		⊕
C.G. of Water Plane __ of ⊠		⊕
Trans. B. M.		
C.B. Above Base Line		⊕ #
Trans. "M" Above Base Line		⊕
C.G. Above Base Line	17.10	#
Trans. "G.M." Full Load Condition		
Longtl. "B.M."		
C.B. Above Base Line		
Longtl. "M" Above Base Line		
C.G. Above Base Line	17.10	
Longtl. "G.M." Full Load Condition		
Foot Tons M to Trim 1"		⊕
Longtl. C.B. __ of ⊠		⊕
Incr. in Displt. per Foot Trim by Stern		⊕

⊠) = (Trans. Mom. of Inertia) ÷ (Cu. Ft. Displ.) = Trans. B. M.

÷ (Cu. Ft. Displ.) = Longl. B. M.

Calculations made on Separate Sheet

⊕ Usually Plotted on "Curves of Form"

Stern = $\frac{\text{Tons per Inch} \times 12 \times \text{C.G. of W.P. Aft of ⊠}}{\text{L.W.L.}}$

ps) = $\frac{\times 12 \times}{415}$ =

M. = $\frac{\times}{12 \times}$ =

this Formula

Effective Length 415'-0"

Curves of Form Cal

Longl. Interval $= \dfrac{\text{Effective Length}}{20} =$ ______

$\dfrac{\text{L.I.} \times 2}{3} =$ Multiplier

$\text{Sum}_1 \times$ Multipr. = Cu. Ft. Mld. Displ.

Sum_2 = Diff. of Mom.

$\dfrac{\text{Sum}_2}{\text{Sum}_1} \times$ L.I. = L.C.B

$\dfrac{\text{Sum}_3 \times 2 \times \text{L.I.}}{3}$ = Area W. P.

$\dfrac{\text{Area of W.P.}}{420}$ = T. P. I.

$\dfrac{\text{Sum}_4 \times}{\text{Sum}}$

$\dfrac{\text{Sum}_5 \times}{3}$

Mom. of

Column No.	DISPLACEMENT (1) Sta.	(2) Plan't'r Reading	Half Area	(3) S.M.	(4) Products	LONGL. C. B. (5) L.L.	(6) Moments	TONS PER INCH (7) ½ Breadth of W.L.	(3) S.M.	(8) Products	(5) L.L.	H.
	0	000						0 0.0				
	1	155						8.2				
	2	342						1 6.5				
	3	493						2 2.6				
	4	574						2 5.5				
	5	612						2 6.6				
	6	626						2 7.0				
	7	626						2 7.0				
	8	626						2 7.0				
	9	626						2 7.0				
	10	626				Ford.		2 7.0			Ford.	
	11	626						2 7.0				
	12	626						2 7.0				
	13	618						2 7.0				
	14	602						2 7.0				
	15	565						2 7.0				
	16	503						2 6.8				
	17	414						2 6.0				
	18	289						2 2.7				
	19	139						1 7.3				
	20	000						0 0.0				
			$\text{Sum}_1 =$			Aft		$\text{Sum}_3 =$			Aft	

X Multiplier ______

= ______ Cu. Ft. Displ.

÷ 35

= ______ S.W. Tons Displ.

Sum_2

$\dfrac{\text{Sum}_2 \times \text{L.I.}}{\text{Sum}_1} = \dfrac{\quad \times \quad}{\quad}$

= ______ Longl. C. B. of ⊠

Ford or Aft

$\dfrac{\text{Sum}_3 \times 2 \times \text{L.I.}}{3} =$ ______ (Area W. P.)

$\dfrac{\text{Area of W.P.}}{420} =$ ______ (Tons per Inch)

Sum_4

$\dfrac{\text{Sum}_4 \times \text{L.I.}}{\text{Sum}_3} =$

= ______

(Ford or Af

Plan'r. Constant $= \dfrac{\text{Half Beam} \times \text{Draft}}{\text{Reading for ⊠ Half Rect.}} = \dfrac{27 \times 24}{636} = \underline{1.0176}$

$\dfrac{\text{Area W.P}}{\text{B.} \times \text{L.}}$ = Water Plane Coeff. = ______

Block Coeff $= \dfrac{\text{Cu. Ft. Displ.}}{\text{L.} \times \text{B.} \times \text{d.}} = \dfrac{\quad}{\quad \times \quad \times \quad} = 0.$

Increase in D

(If C.G. of W.

(Substitute "

Max. Section Coeff. $= \dfrac{2 \times \text{Max. Half Area}}{\text{B.} \times \text{d.}} = \dfrac{2 \times}{\quad \times} = \underline{0.}$

Prismatic Coeff. $= \dfrac{\text{Block Coeff.}}{\text{Max. Sec. Coeff}} = \dfrac{0.}{0.} = \underline{0.}$

! Mom. to Tri

! B.M. is Usua

Fig. 4.—Curve

...culations for (Example Ship)

To 24'-0" W.L.

$\frac{\ldots L.I.}{3}$ = C.G. of W.P. Abt. ⊠

$\frac{\ldots L.I.^3 \times 2}{3}$ = Mom. of Inertia of W.P. Abt. ⊠

$\frac{I - (\text{Area W.L.} \times \text{C.G.})^2}{\text{Cu. Ft. Displ.}}$ = Longl. B. M.

$\frac{Sum_6 \times 2 \times L.I.}{9}$ = Trans. Mom. of Inertia

$\frac{\text{Mom. of Inertia}}{\text{Cu. Ft. Displ.}}$ = Trans. B.M.

LONGL. B. M.			TRANS. B. M.		
(9)	(5)	(10)	(11)	(3)	(12)
...B. x L. L.	L.L.	H.B. x (L.L.)^2	$\frac{1}{2}$ B. of W.^{3}	S.M.	Products
	Sum$_5$		Sum$_6$ =		

SUMMARY		
Molded Displacement S.W.		
Shell & Appendg. Displ. (6%)		#
Total S. W. Displacement		⊕
Block Coeff.		⊕
Max. Section Coeff.		⊕
Prismatic Coeff.		⊕
Water Plane Coeff.		⊕
Tons per Inch S. W.		⊕
C. G. of Water Plane __of ⊠		⊕
Trans. B. M.		⊕ }
C.B. Above Base Line		# }
Trans. "M" Above Base Line		⊕
C.G. Above Base Line	1 7 . 1 0	#
Trans. "G. M." Full Load Condition		
Longtl. "B. M."		
C. B. Above Base Line		
Longtl. "M" Above Base Line		
C.G. Above Base Line	1 7 . 1 0	
Longtl. "G. M." Full Load Condition		
Foot Tons M to Trim 1"		⊕
Longtl. C. B. __of ⊠		⊕
Incr. in Displt. per Foot Trim by Stern		⊕

x

C.G. of W.P. __ of ⊠

...t)

$\frac{Sum_5 \times 2 \times L.I.^3}{3} = \_\_ \times \frac{2}{3} \times \_\_$

$\frac{Sum_6 \times 2 \times L.I.}{9} = \_\_ \times \frac{2}{9} \times \_\_$

= $\overline{\text{(Mom. of Inertia Abt. ⊠)}}$

= $\overline{\text{(Trans. Mom. of Inertia)}} \div \overline{\text{(Cu. Ft. Displ.)}}$ = $\overline{\text{Trans. B. M.}}$

− $\overline{(\text{Area W. P.} \times \text{C.G.}^2)}$

= $\overline{\left(\begin{array}{l}\text{Mom. of Inertia}\\ \text{Abt. C.G. of W.P.}\end{array}\right)} \div \overline{\text{(Cu. Ft. Displ.)}}$ = $\overline{\text{Longl. B. M.}}$

\# Calculations made on Separate Sheet

⊕ Usually Plotted on "Curves of Form"

$\frac{}{\ldots x}$ = 0.

...Displacement per Foot of Trim by Stern = $\frac{\text{Tons per Inch} \times 12 \times \text{C.G. of W. P. Aft of ⊠}}{\text{L. W. L.}}$

...P. should be Forward of Midships)
... Decrease " for Increase

$= \frac{\times 12 \times}{415} =$

...m 1" = $\frac{\text{Tons Displ.} \times \text{Longl. G. M.}}{12 \times \text{L.W.L.}} = \frac{x}{12\ x} =$

...lly Substituted for G. M. in this Formula

...es of form calculations

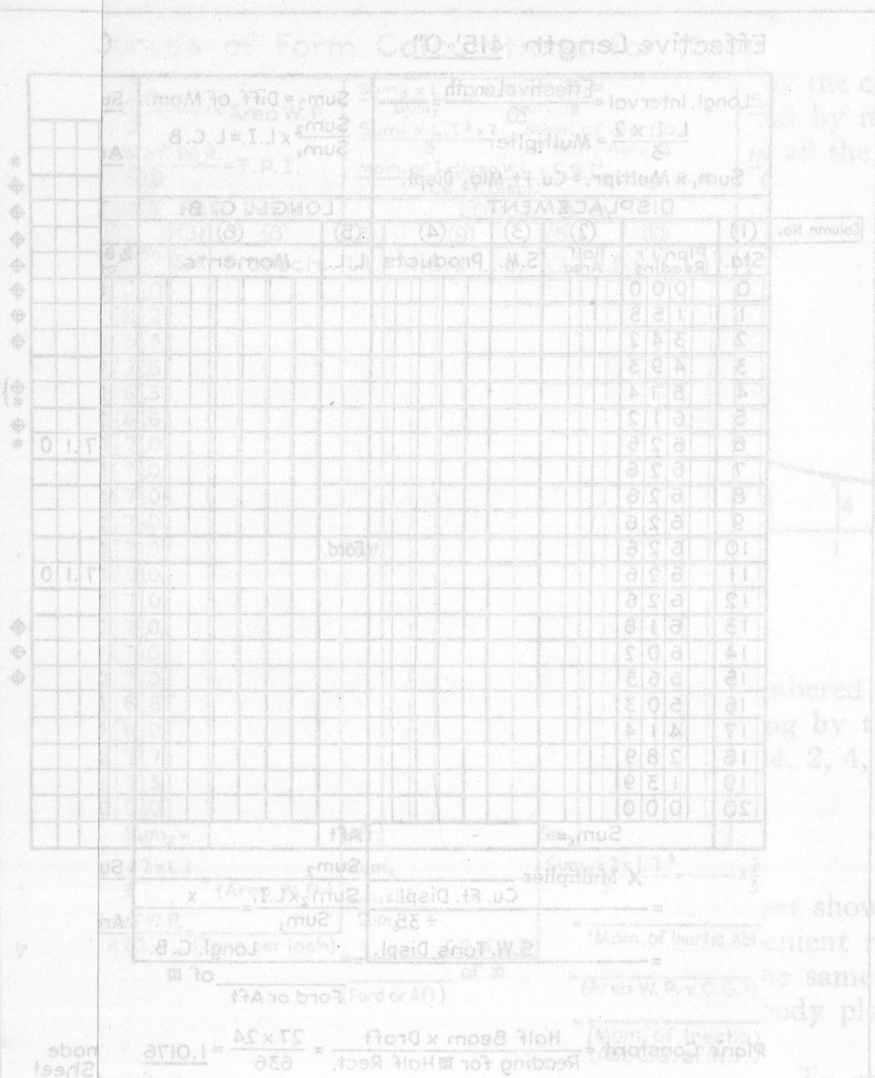


Fig. 4.—Curves of form calculations

the scale of the drawing, but which can always be determined by running the point around a known area.

In Fig. 4 the planimeter readings for the body plan shown in Fig. 1, up to the 24-foot waterline, are assumed to have been obtained and tabulated as shown. The half-areas, which are the product of the readings and the planimeter constant, should be entered in column 2. The proper Simpson's multipliers are entered in column 3 (this column is repeated after column 7 and after column 11, for convenience). The product of each half-area and its multiplier is entered in column 4 and the sum of these products written at the foot of the column as "Sum_1." This sum, multiplied by the distance between stations (the "longitudinal interval"), by 2 for both sides of the ship, and divided by 3 as required by Simpson's rule, will give the displacement in cubic feet. The factors comprising this multiplier are multiplied together and the product written on the calculation sheet as "multiplier" just below the sum of column 4. Dividing the displacement in cubic feet by 35, the number of cubic feet of salt water in a ton, gives the displacement in tons of salt water.

The displacement obtained as in the foregoing is the displacement of the molded form; the total displacement will be more than this by the displacement of the shell plating and the appendages. The shell displacement is usually taken as a percentage (0.6 percent at load draft up to 1 percent at very light draft) of the molded displacement. In special cases, such as an armored ship, or one with a heavy ice belt, the shell displacement must actually be calculated. The volume of the appendages, namely, the rudder, propellers, shafts, shaft struts or bossings, if there are any, the bilge keels, etc., must be calculated by any means that is most convenient. The sum of all these is added to the molded displacement to get the total displacement, as is shown in the summary at the right-hand edge of the calculation sheet. For a single-screw ship the appendages are small (the rudder can be considered as replacing the aperture in which the propeller is placed) and the total shell and

appendage displacement may be taken as about 0.6 to 1 percent of the molded displacement. In Fig. 4, 0.6 percent is used.

Longitudinal Center of Buoyancy (Columns 5 and 6). To find the center of buoyancy, or center of gravity, of the displacement, we must find the moment of the displacement about some point, and divide this moment by the displacement. This will give the distance from the center of buoyancy to that point, as will be seen from the definition of a center of gravity.

If each of the products in column 4 is multiplied by its distance from amidships (station 10), the new product is a function of the moment of the area of each station about amidship. Instead of using the actual distance in feet, the number of stations from amidships to each station will be used, and the length in feet of the interval between stations combined with the multiplier.

Write in column 5 the longitudinal lever; that is, the number of stations from each station to amidships. Multiply column 4 by column 5 and write the products in column 6, add separately the forward and after products, subtract whichever is smaller from the other and write the remainder as the "Sum_2" at the bottom of column 6. If the forward sum is greater than the after sum, the final sum will be forward, and vice versa.

Written out in full, the moment of the displacement about amidship would be

$$\text{Sum}_2 \times \tfrac{2}{3}\, LI \times LI$$

The displacement is

$$\text{Sum}_1 \times \tfrac{2}{3}\, LI$$

Therefore the *LCB*, which is moment of displacement divided by displacement, is

$$\frac{\text{Sum}_2 \times \frac{2}{3}\, LI \times LI}{\text{Sum}_1 \times \frac{2}{3}\, LI}$$

or cancelling out those terms which are in both numerator and denominator,

$$LCB = \frac{\text{Sum}_2 \times LI}{\text{Sum}_1}$$

Suitable spaces are provided on the calculation sheet for this operation.

If the ship is to float at the designed waterline, not only must the weight of the ship equal the displacement to that waterline, but the *LCB* must be the same distance forward or aft of amidships as is the center of gravity of the ship's weight.

Tons per Inch of Immersion (Columns 7 and 8). The tons per inch is the displacement, or weight, of a layer of water 1 inch thick at any given waterline. The volume of this 1-inch layer in cubic feet is the area of the waterplane in square feet multiplied by its thickness, $\frac{1}{12}$ foot. Its displacement in tons will be its volume in cubic feet divided by 35, the number of cubic feet of salt water per ton. So the tons per inch at a given waterline is simply the area of the waterplane divided by (12×35) or 420.

If a known weight is to be added to or taken from the ship, we can find the resulting change of mean draft, in inches, by dividing this weight by the tons per inch at the draft at which the ship is floating.

Columns 7 and 8 of Fig. 4 give the area of waterplane. The half-breadths of the waterline at each station are put in column 7, multiplied by Simpson's multipliers, the products entered in column 8 and added to get "Sum_3." This sum, multiplied by $\frac{2}{3}$ the longitudinal interval will give the area of waterplane; this area divided by 420 will give the tons per inch.

Metacentric Radius (*BM*) (Columns 9 to 12). The distance from the center of buoyancy to the metacenter, usually called *BM*, is really the radius of curvature of the line along which the center of buoyancy moves as the ship is inclined. It is therefore called the metacentric radius.

It can be proved [see Reference (*a*), page 105] that the

metacentric radius equals the moment of inertia of the waterplane divided by the volume of the displacement. That is

$$BM = I/V$$

This is an important equation and should be memorized.

The moment of inertia of the waterplane, which is more easily used expressed in terms of metacentric radius, is the quality which keeps a surface vessel (not a submarine) from capsizing. It will be seen in Chapter III how this quantity controls stability by keeping M a satisfactory distance above G.

For longitudinal BM the longitudinal moment of inertia of waterplane is used; that is, the moment of inertia about a transverse axis through the center of gravity of the waterplane. For transverse BM the transverse moment of inertia is used; that is, the moment of inertia about a fore-and-aft axis through its center of gravity (which will lie on the centerline of the ship). The volume of displacement V is, of course, the same for both longitudinal and transverse BM.

Longitudinal *BM* (Columns 9 and 10). Before we can calculate this, we must find the longitudinal moment of inertia of the waterplane about its LCG. This in turn requires us to find the LCG of the waterplane.

Aside from its use in getting longitudinal BM, the center of gravity of the waterplane is of interest because it is the point at which an added weight must be placed to result in no change in trim, and the point about which the ship will rotate, or change trim, when a weight already on board is shifted in a fore-and-aft direction. For this reason the CG of the waterplane is called the center of flotation.

The LCG of waterplane is obtained in column 9 from column 8 in exactly the same way as LCB was obtained in column 6 from column 4. Multiply the products of the half-breadths of waterline and SM, which are already tabulated in column 8, by the longitudinal lever, again expressed in intervals instead of feet, and write in column 9 the resulting functions of the moments of the half-breadths of the water-

line about amidship. Find the excess forward or after moment. This final moment, "Sum_4," divided by "Sum_3" and multiplied by the longitudinal interval, will give the *LCG* of waterplane.

To get the longitudinal moment of inertia of waterplane about amidships, apply Simpson's rule to the product of each half-breadth and the *square* of its distance from amidships. We already have in column 9 the product of each half-breadth and its distance from amidships expressed in intervals instead of in feet. If we multiply each item in column 9 by its longitudinal lever once more, we will have the desired products. These are entered in column 10. Since the square of either a positive or negative number is positive, these products in column 10 are added directly, regardless of whether they are forward or aft of amidships. The "Sum_5" multiplied by ⅔ times the longitudinal interval *cubed* (once for Simpson's rule and twice because we have twice used the longitudinal lever in intervals instead of in feet) gives the longitudinal moment of inertia of the waterplane about amidships. This is not yet quite what we need; we need the moment of inertia about *LCG* of waterplane instead of about amidships. The moment of inertia of an area about an axis not through the *CG* of the area is equal to the moment of inertia about an axis through the *CG* plus the product of the area times the square of the distance from the *CG* to the axis. Hence the moment of inertia we have just obtained is too large, by an amount equal to the area of waterplane times the square of the distance from amidships to the *LCG* of *WP*. We have already obtained both of these quantities, so the correction required to change the moment of inertia about amidships to moment of inertia about the *CG* of the waterplane can be calculated. Appropriate spaces will be found in Fig. 4. Longitudinal *BM* may now be found by dividing the moment of inertia about *CG* of waterplane by the cubic feet of displacement, found at the bottom of column 4.

Transverse *BM* (Columns 11 and 12). The transverse moment of inertia of the waterplane can be obtained more

simply than was the longitudinal moment of inertia, by using the fact that the moment of inertia of a rectangle about one edge is ($LB^3 \div 3$) where L is the length of the edge used as an axis and B is the length at right angles to the axis.

We will treat each half-breadth as the B of a rectangle whose length is the longitudinal interval. Write the cube of each half-breadth in column 11, using, if convenient, a table of cubes instead of multiplying them out. Multiply by Simpson's multipliers and write the products in column 12. Add column 12 to get "Sum_6." Multiply "Sum_6" by $\frac{1}{3}$ for Simpson's rule, by $\frac{1}{3}$ again for the inertia formula previously quoted and by 2 for both sides of the ship; that is, by $\frac{2}{9}$ in all, and get the transverse moment of inertia of the waterplane. Divide this by the volume of the displacement from the foot of column 4 and get the transverse BM.

Vertical Center of Buoyancy. The summary on the form calculation sheet shows just below the line for "transverse BM," a line for "CB above the base line" or VCB. This, added to transverse BM, gives the height of the transverse metacenter above the keel, KM.

VCB is calculated on a separate sheet, and must wait until the displacement has been calculated at a sufficient number of waterlines to permit drawing a complete curve of displacement plotted against draft.

It can be shown [see Reference (*b*), page 68, for a start] that, if, in Fig. 5, A is the area below the displacement curve up to any given waterline, and B the entire area of rectangle drawn as shown, then

$$VCB = \text{draft} \times \frac{\text{area } A}{\text{area } B}$$

After the displacement curve has been drawn in Fig. 7, we can, at each waterline used in the calculations, measure with the planimeter the areas A and B. Since only the ratio of these areas is needed, the planimeter readings can be used directly, without using any constant. Then for each

waterline multiply the draft by the reading for *A* divided by the reading for *B*. This ratio of *A* to *B* will not vary much and will be in the neighborhood of 0.54. This will give the *VCB* for each draft used.

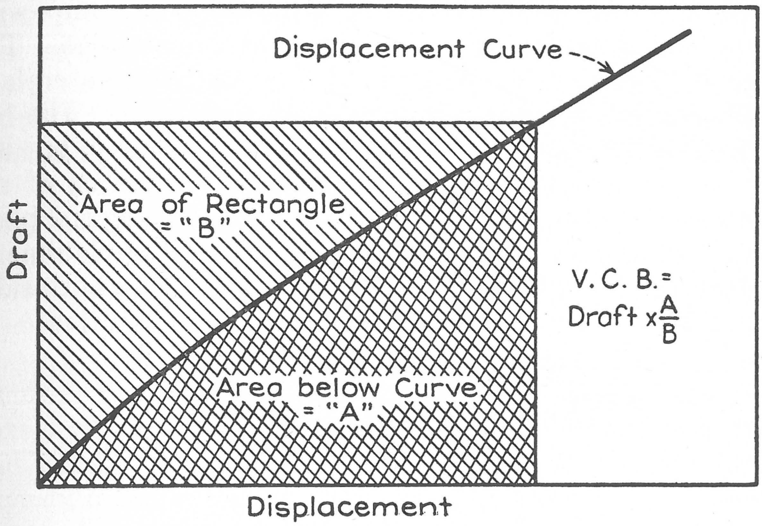


Fig. 5

One other line on the summary is not calculated on this sheet, the "*CG* above base line," or *VCG*. This calculation, described in Chapter VI, consists of listing all the weights comprising the ship and its load, together with the height of the *CG* of each item above the base line. Each item of weight is multiplied by its *VCG* and the sum of all the resulting moments divided by the sum of all the weights. The quotient is the *VCG* of the ship and its load. The *LCG* is usually calculated at the same time.

In preliminary work *VCG* is usually proportioned from similar ships on which it has been calculated, or on which it has been determined by an inclining experiment.

When the *VCG* is subtracted from the "Transverse metacenter above *BL*," or *KM*, the difference is the distance from

the CG to the transverse metacenter. *This distance is called GM, or metacentric height, and controls the initial transverse stability of the ship, as will be shown in Chapter III.*

Form Coefficients. The form calculation sheet shows places for calculating several "coefficients of fineness." These coefficients are useful in comparing one ship with another of different size, and in power calculations. The method of getting the block coefficient and the waterplane coefficient follows directly from their definition. The numerator of the maximum section coefficient (usually, but not always, the midship section coefficient) is the area of the maximum section, and is obtained by multiplying the planimeter reading for the maximum section by the planimeter constant to change it to square feet, and by 2 for both sides of the ship. The prismatic coefficient is by definition:

$$C_p = \frac{\text{volume of displacement}}{\text{length} \times (\text{area of maximum section})}$$

or

$$C_p = \frac{L \times B \times D \times \text{block coefficient}}{L \times (B \times D \times \text{maximum section coefficient})}$$

which reduces to

$$C_p = \frac{\text{block coefficient}}{\text{maximum section coefficient}}$$

Moment to Trim One Inch. Instead of using the longitudinal metacentric height directly to investigate the trim resulting from a fore-and-aft trimming moment, it is more convenient to calculate the moment required to change the trim of the vessel one inch. Then, if we need to determine the change of trim that would result from any trimming moment, due either to a fore-and-aft movement of a weight or to a trimming lever (the distance between the LCB on even keel and the LCB resulting from a given loading), the trim may be found by dividing the trimming moment by the moment to trim one inch.

The moment to trim one inch can be shown to be [see Reference (*b*), page 155]:

$$\text{Moment to trim 1 inch} = \frac{\text{displacement} \times \text{longitudinal } GM}{12 \times L}$$

Since the longitudinal *GM* is not known until *VCG* is known and will vary with each condition of loading, it is usual to use longitudinal *BM* instead. The error involved is small.

Increase in Displacement per Foot of Trim by the Stern. The displacement has been calculated with the ship on an even keel. If the ship is not on an even keel, but is trimmed by the bow or by the stern, and we wish to read the displacement from the displacement curve (see curves of form, following), we use the mean draft, which is the average of the bow and stern drafts, or the draft amidships. But the ship does not necessarily have the same displacement as it would if it were floating at the same mean draft on an even keel. If the *CG* of *WP* is aft of amidships (as it usually is) and the ship is trimmed by the stern, the displacement read from the curve must be increased for each foot of trim by a correction equal to [see Reference (*b*), page 153]:

$$\frac{\text{tons per inch} \times 12 \times (CG \text{ of } WP \text{ aft of amidships})}{LWL}$$

If the *CG* of *WP* is forward of amidships, the correction must be subtracted from the displacement read from the curve.

This correction is approximate and is used for moderate amounts of trim, up to perhaps 2 percent of the ship's length. For excessive amounts of trim, an accurate displacement can be obtained only by making an independent displacement calculation carrying each station up to the inclined waterline (see "Bonjean Curves" following).

Curves of Form. After the various form characteristics have been calculated on the form calculation sheet for a number of drafts covering a range from the least expected for the empty ship to the greatest expected for the fully loaded ship (one calculation sheet for each draft), a drawing is

made called "Curves of Form" (see Fig. 7). A vertical scale for draft is chosen to suit the size of drawing desired. Then, for displacement, for example, each calculated value of displacement is laid off, measuring from the left-hand edge of the sheet, at its appropriate draft, and a smooth curve drawn through all these calculated points. From this curve may be read the displacement at any other draft between the calculated points. Similar curves are drawn for all the form characteristics indicated in the summary. Sometimes in preliminary work only the most used ones will be plotted, as displacement, transverse *KM* and *LCG*.

Bonjean Curves. If it is necessary to find the displacement of a ship up to a waterline having an excessive trim, as during launching, or up to an irregular waterline, such as the surface of the large wave used in standard strength calculations, we will need "Bonjean Curves" (see Fig. 6).

Lay down a base line representing the length of the ship, and erect a perpendicular at each station. A total length of 40 inches, with the stations 2 inches apart, is customary. A vertical scale of draft is chosen to give the desired height of drawing. Then at any given station, at the proper height, lay off to the right, from the station line as zero, each calculated area. A smooth curve is drawn through these points, and from this curve we may read the area of the station at any intermediate height. A similar curve is drawn for each station.

Any desired waterline may now be drawn across this diagram and the areas of each station up to this waterline can be read from the Bonjean curves at the height at which the waterline crosses the station. From these areas, the displacement and *LCB* may be calculated as previously described.

Approximations. The following approximations are useful both to check calculated figures for error and to estimate probable values without making calculations.

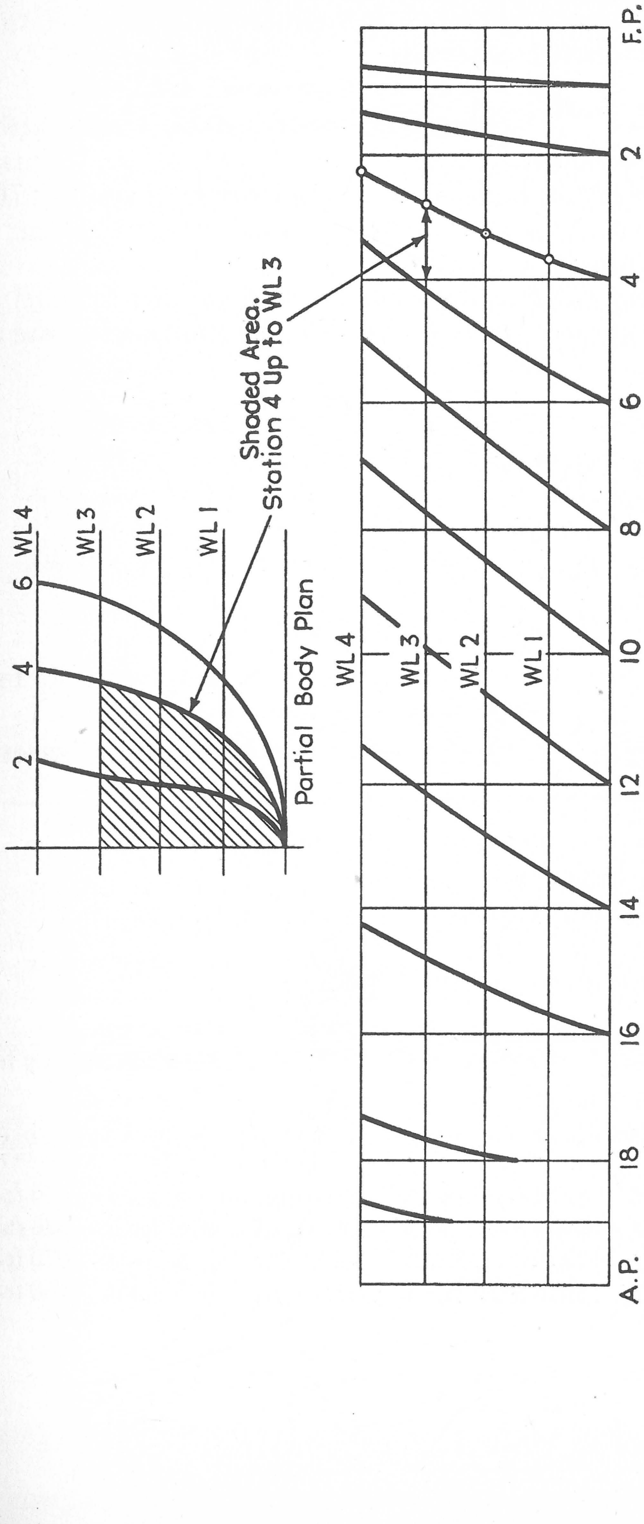


Fig. 6.—Bonjean curves. The area of station 4, for example, up to *WL*3, is plotted as indicated by the arrow. The area of this station up to the other waterlines is similarly plotted, and the Bonjean curve for station 4 drawn through these points. This is done with each station.

KB will vary from 0.52 to 0.56 times draft. A very close approximation is given by

$$KB = \text{draft} \times \frac{\text{waterplane coefficient}}{\text{waterplane coefficient} + \text{block coefficient}}$$

BM = approximately 0.08 $(\text{beam})^2/\text{draft}$. This is a close approximation for block coefficients of about 0.70, but it can vary from about 0.075 for fine vessels to about 0.085 for full vessels.

Moment to trim an inch is approximately $(L/100)^2 \times B$. A closer approximation is obtained by multiplying the above by a factor as follows:

If waterplane coefficient is 0.60, use factor of 0.87.
If waterplane coefficient is 0.65, use factor of 0.97.
If waterplane coefficient is 0.70, use factor of 1.09.
If waterplane coefficient is 0.75, use factor of 1.23.
If waterplane coefficient is 0.80, use factor of 1.40.

Problems

1. Fill out "Form Calculation Sheet," Fig. 4, for the example ship, for 24-foot waterline.

2. Draw "Curves of Form" on cross-section sheet, Fig. 7, showing:

(a) Total displacement in salt water.
(b) Metacenter above *BL*.
(c) Tons per inch.
(d) *LCB*.
(e) Moment to trim 1 inch.

using, in addition to the calculated values for the 24-foot waterline obtained in Problem 1, the following values for the lower waterlines:

	(a)	(b)	(c)	(d)	(e)
8′ *WL*	3370 tons	33.7′	40.4 tons	11.5′ for'd	860 ft-tons
12′ *WL*	5370 tons	26.2′	42.1 tons	9.4′ for'd	970 ft-tons
16′ *WL*	7440 tons	23.6′	43.7 tons	7.5′ for'd	1070 ft-tons
20′ *WL*	9570 tons	22.7′	45.1 tons	5.7′ for'd	1160 ft-tons

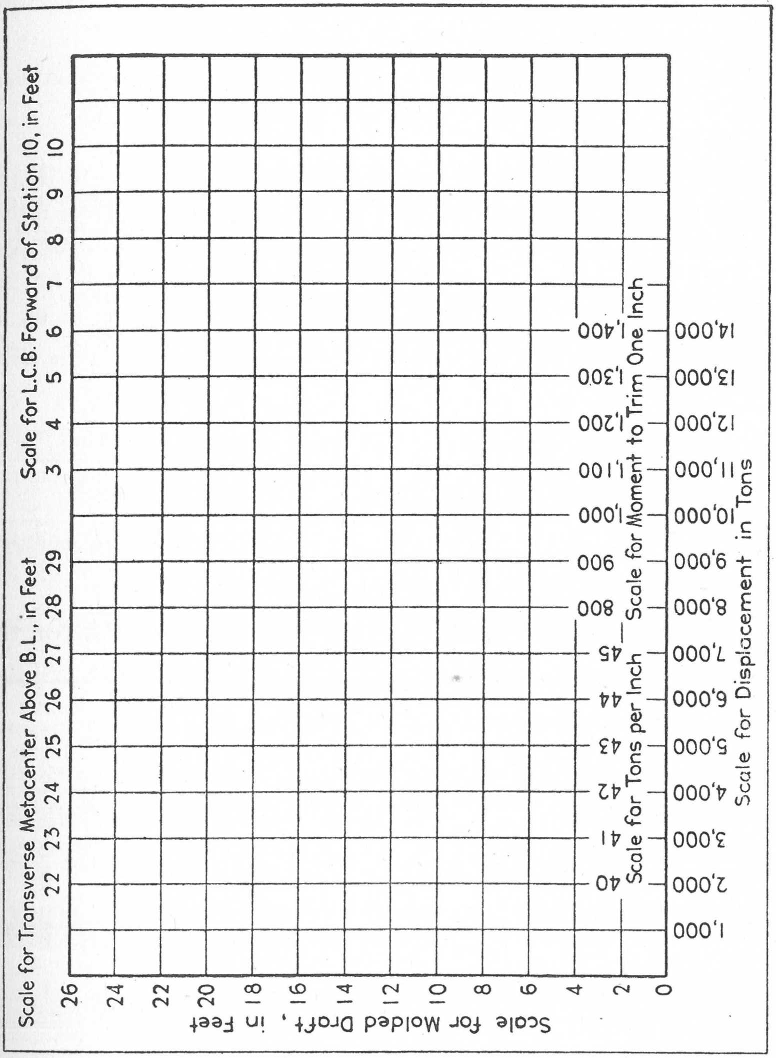

Scale for Transverse Metacenter Above B.L., in Feet
22 23 24 25 26 27 28 29
Scale for L.C.B. Forward of Station 10, in Feet
3 4 5 6 7 8 9 10
Scale for Molded Draft, in Feet
0 2 4 6 8 10 12 14 16 18 20 22 24 26
Scale for Tons per Inch
40 41 42 43 44 45
Scale for Moment to Trim One Inch
800 900 1,000 1,100 1,200 1,300 1,400
Scale for Displacement in Tons
1,000 2,000 3,000 4,000 5,000 6,000 7,000 8,000 9,000 10,000 11,000 12,000 13,000 14,000

Fig. 7

3. If the example ship is loaded to 10,500 tons with a *VCG* of 20.1 feet and a *LCG* of 6 feet forward of amidships, using the curves of form drawn in Problem 2, what will be her

(a) mean draft (neglect correction for trim).
(b) *GM*.
(c) trim..

4. A certain ship is allowed to load to 20 feet draft, or to a displacement of 10,000 tons, in salt water (35 cubic feet per ton). She is loading in fresh water of 36 cubic feet per ton. Her "tons-per-inch" is 50 tons at and near 20-foot draft. How deep may she load at the pier so as to draw 20 feet at sea? (Neglect consumption of fuel, etc., proceeding to sea.)

Starter: Volume of displacement at sea = 10,000 × 35 = 350,000 cubic feet
Volume of displacement at pier = 10,000 × 36 = 360,000 cubic feet
Excess volume of displacement at pier = 10,000 cubic feet

5. Find the difference in height of *M* above *BL* for the rectangular and triangular vessels shown in Fig. 8. Length for both is 50 feet.

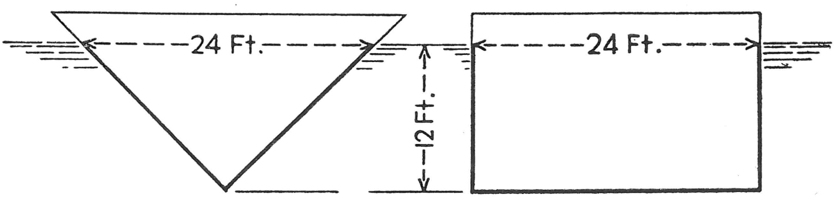


Fig. 8

Starter: Find *VCB* of each form.
I of waterplane is same for both.
I/*V* is greater for the triangular form.
Length is immaterial.

6. The example ship must cross a certain bar where there is only 12 feet of water. The ship is drawing 10 feet 8 inches forward, 12 feet 4 inches aft, 11 feet 6 inches mean. The forepeak tank, which is empty, holds 90 tons salt water ballast, with its center 200 feet forward of the center of flotation. Can the vessel be ballasted so as to cross this bar by filling or partially filling the forepeak?

Starter: Find tons per inch and moment to trim 1 inch from curves of form. Find effect of filling forepeak on mean draft and on trim, and get resulting drafts.

Chapter III

STABILITY, INCLINING EXPERIMENT AND PERIOD OF ROLL

References

(*a*) "Principles of Naval Architecture," Volume I, Chapters III and IV.

(*b*) Attwood's "Theoretical Naval Architecture," Chapters III and V.

(*c*) Peabody's "Naval Architecture."

Stability

Stability is the tendency of a ship to return to the original position when inclined away from that position.

In the case of a ship floating upright, or nearly upright, this tendency depends on the amount of GM, or metacentric height; that is, the vertical distance between the center of gravity and the metacenter.

Fig. 9 shows a ship inclined transversely. (If the sketch were a profile, showing a ship inclined longitudinally, exactly the same reasoning would apply, but the longitudinal M would be much higher.) Note that M, the transverse metacenter, might be described as *the point that the center of buoyancy stays under.* The weight of the ship acts down, through G, and the buoyancy acts up, through B'. These two equal and opposite forces constitute a couple, whose moment is equal to either force times the distance between them; that is, to $\Delta \times GZ$. GZ is called the righting lever, and, at a small angle of inclination, equals $GM \sin \theta$. The righting moment $= \Delta \times GM \times \sin \theta$, and is directly proportional to the metacentric height, GM. If a heeling moment is applied to a ship, the resulting angle of heel will be such that the righting moment equals the heeling moment.

If the heeling moment is $W \times d$ (a weight W, d feet from centerline), then $Wd = \Delta GM \sin \theta$, and $\sin \theta = Wd/\Delta GM$. We will refer to this formula later under "Inclining Experiment."

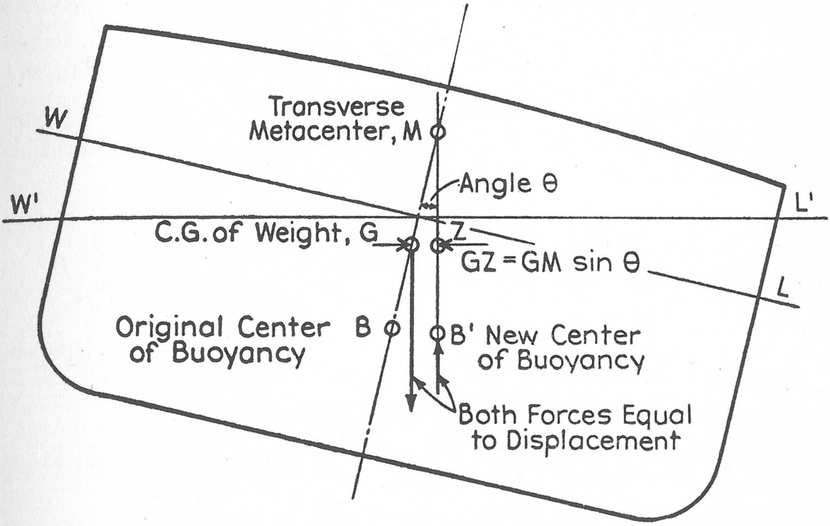


Fig. 9

As far as stability is concerned, the effect of the buoyancy is as though it were at M instead of at B'. If M is above G, the moment of weight and buoyancy tends to right the ship and the ship is stable. If M is at G, GZ is zero, the couple does not exist and the ship is in neutral equilibrium. If M is below G, the couple tends to capsize the ship and the ship is unstable. Also, if GM is small, the righting moment is small and the ship will return to the upright slowly, with a long period of roll. If GM is large, the roll will be relatively quick. Instability is dangerous, so GM must not be too small. Quick rolling is undesirable, so GM must not be too large. Obviously, M is an important point.

The calculation of the height of B and M is explained under "Form Calculations," Chapter II, and the height of G under "Weights," Chapter VI.

Rocking-Chair Analogy. It may help to visualize the equilibrium of a ship by comparing it to an ordinary rocking-chair (see Fig. 10).

G is the combined center of gravity of chair and occupant.

B is the point of contact with the floor, or point of support.

M is the center of curvature of the rocker and is the "metacenter" of the chair, or the point that *B* stays under as the chair rocks.

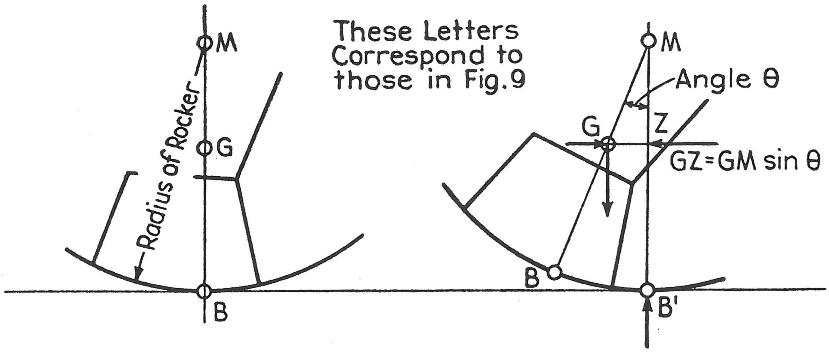


Fig. 10

B corresponds to the "center of buoyancy."

BM corresponds to the "metacentric radius."

GM corresponds to the "metacentric height."

GZ corresponds to the "righting lever."

M must be above *G* for the chair to be stable. If one stands on the chair and grasps the back, so as to bring the combined *G* higher than *M*, the chair will capsize. Notice that the back end of the rocker is usually made flatter than the rest; that is, its center of curvature, or *M*, is higher, so that, if the chair is inadvertently rocked too far, *M* rises sharply and tends to prevent a disaster. This we cannot do on a ship, except that in a small way the sharp flare or sponson just under the deck of a river or bay steamer has a similar effect.

This analogy persists regarding the effect of GM on period of roll. The empty chair will have a low G and a high GM, and will rock quickly. An occupied chair will have a higher combined G, a lower GM, and will rock more slowly. If the occupant slowly raises himself in the chair, raising G, the period of roll will lengthen until when G almost reaches M it will rock very slowly, having but little tendency to return.

A ship behaves just so. A high GM results in a quick roll and a low GM results in a slow roll.

This rocking-chair stability can all be demonstrated at home, and helps one to realize why the ship behaves as it does.

For ordinary ship forms BM has been found to be equal to B^2/d times a factor. This formula shows that beam is the most effective factor controlling BM, and, with it, GM. GM can be increased somewhat by filling the ends of the waterline and so increasing I, but any substantial change in GM must be made by changing beam. *In design, therefore, the beam is determined by the required GM.* This required beam may be wider than is otherwise desirable; resistance, for instance, generally increases with beam-draft ratio, but stability takes precedence over low resistance.

Free Surface. We have seen that, as far as stability is concerned, the effect of buoyancy of a ship is as though it were at M instead of at B. In exactly the same way, the effect of the weight of a liquid in a tank (Fig. 11) is as though it were at its own metacenter instead of at its center of gravity. This effect can be realized by holding a shallow pan of water in the hands; the weight feels as though the water were at a considerable height above the pan. The height of the liquid's own metacenter above the center of gravity of the liquid (gm in the figure) can be proved to be

$$gm = i/v$$

where i = moment of inertia of the free surface of the liquid about its own center of gravity, and v = volume of the liquid.

Note the similarity between this formula and that for BM of a ship: $BM = I/V$. Note also that, if a tank is completely filled, there is no free surface, i becomes zero, gm is zero and m is at g.

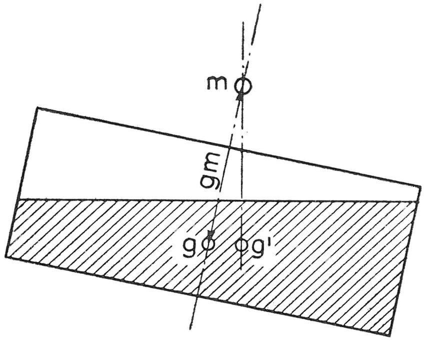


Fig. 11

This effect must be considered when calculating the center of gravity of a ship with liquids in her tanks. It can be done in either of two ways: (1) calculate the m (sometimes called the virtual center of gravity) of the liquid in each tank and consider the weight of the liquid to be at these points, or (2) consider the liquid to be at its actual center of gravity, calculate and add together all the moments of inertia of all the free surfaces, and obtain a single correction to the ship's GM by the formula:

$$\text{Free surface correction to } GM = \frac{\text{sum of } i \text{ of tanks}}{\text{volume of displacement of ship}}$$

For proof that these methods are equivalent, see Reference (*b*), page 131.

If the liquid in the tank has a specific gravity different from that of the water in which the ship is floating, i must be multiplied by the ratio

$$\frac{\text{Specific gravity of liquid in tank}}{\text{Specific gravity of water in which ship is floating}}$$

before calculating the above free surface correction.

Usually it is sufficiently accurate to use for i the transverse moment of inertia of a rectangle whose length and width are the mean length and width of the tank. This moment of inertia, about the center of gravity of the rectangle, is $lb^3/12$, where l is the length and b the width of the tank.

Inclining Experiment

We have seen that, if a vessel is inclined, the righting moment of the buoyancy equals $\Delta \times GM \times \sin\theta$. This permits us to find out just what the GM is by inclining the ship by a known inclining moment and measuring the angle of inclination.

The object in determining GM is to locate the VCG. GM is the distance from the VCG to M, the transverse metacenter. The latter can be calculated easily and accurately; but the calculation of the VCG is long and, at best, inaccurate. Accordingly, it is usual, and for passenger ships compulsory, to locate VCG definitely by an inclining experiment. Then, if VCG is known accurately in the "as inclined" condition of loading, one can calculate VCG and GM in any other condition of loading.

The experiment is made as follows: A known inclining moment is applied to the ship by moving a known "inclining weight" across the deck a measured distance. The resulting angle of inclination is measured by a long pendulum or by a micrometer level. When the ship has come to rest after the movement of the weight, the righting moment must equal the inclining moment.

$$\text{Inclining moment} = w \times d = \text{righting moment} = \Delta \times GM \times \sin\theta$$

or

$$GM = \frac{wd}{\Delta \times \sin\theta}$$

where w is the inclining weight, and d the distance through which it is moved. In practice $\tan\theta$ is used instead of sin

θ, because this can be measured directly from the pendulum as

$$\frac{\text{distance pendulum swings}}{\text{length of pendulum}}$$

and because the inclination is kept down to a small angle (2 degrees or so each way) where the sine, the tangent and the actual angle expressed in radians (which theoretically should be used) are for practical purposes all the same.

The chief labor in an inclining experiment consists in determining accurately the condition of the ship as inclined, as to liquids in tanks, their free surface, weights still to go on board to complete the ship, weights which do not belong on board and will be taken off, etc. This is necessary because we start with the weight and *VCG* of the ship "as inclined" (the weight determined by the displacement shown by the draft, and the *VCG* as found by the inclining experiment) and work back to the "light ship, complete," by (on paper) deducting all the items on board which are not part of the light ship, such as liquids, staging, etc., and adding all weights yet to go on board, such as boats, furniture, etc. After calculating the weight and *CG* of the "light ship," the weight, *CG* and resulting *GM* of the ship in any other condition can be calculated.

The difficulty of accurately estimating the free surface of liquids in tanks leads us to eliminate it as far as possible, by either emptying or completely filling, for the inclining experiment, any tanks that have been partly filled.

Stability at Large Angles of Heel

See Reference (*a*), pages 113 to 119; Reference (*b*), pages 174 to 187 and 197 to 120.

The preceding discussion of stability as determined by *GM* fails at large angles because *M* does not stay at one place except for small angles; roughly, angles less than 10 degrees. The stability at large angles of inclination is determined by finding *GZ* (see Fig. 12). This is no longer equal to *GM* sin θ and is found by measuring the actual righting moment

of the displacement, at several angles of inclination, with an integrator. The integrator is an instrument similar in operation to a planimeter, but which measures the moment of an area about an axis as well as the area itself. For description see Reference (*a*), page 22, and for proof see Reference (*c*), page 15.

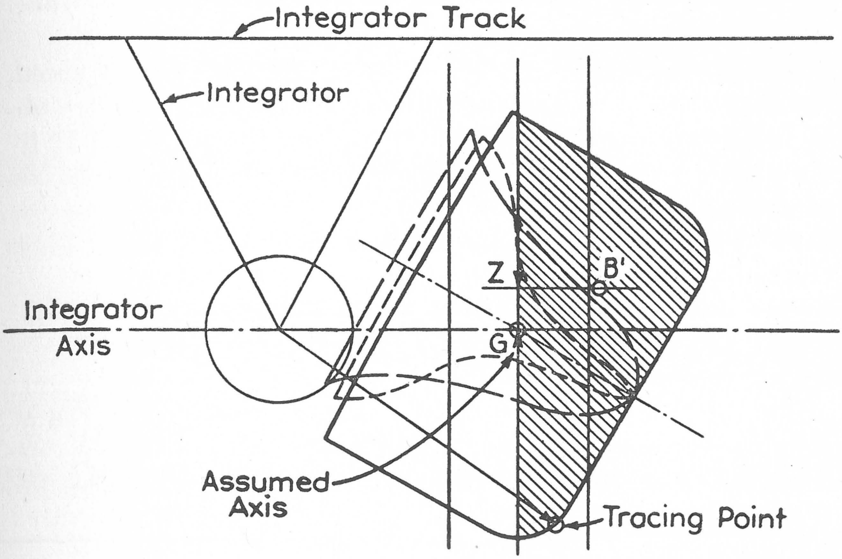


Fig. 12

Curves of *GZ* plotted against angle of heel look about like Fig. 13. For small angles *GZ* equals (*GM* sin θ), but at greater angles the width of the inclined waterplane increases and *GZ* becomes greater than (*GM* sin θ). It reaches a maximum soon after the deck edge becomes immersed, then decreases and finally becomes negative. The shape of the curve depends primarily on *GM* and freeboard. A vessel with low *GM* but high freeboard will have less *GZ* at small angles but more *GZ* at large angles than a similar vessel with high *GM* and low freeboard. Reference (*a*), page 122, and Reference (*b*), page 183, give good discussions of the effect of form on the shape of *GZ* curves.

The steps involved in obtaining GZ with the integrator are as follows (see Fig. 12):

1. A body plan of the ship showing both sides of all stations is constructed and placed at a series of inclinations to the vertical.
2. At each inclination, the displacement of the ship to a series of inclined waterlines is obtained in the usual way (see "Form Calculations," Chapter II), using the area wheel of the integrator instead of a planimeter.
3. At the same time the areas of sections are obtained, the moments of the stations about an assumed axis are obtained from the moment wheel of the integrator, and by Simpson's rule the entire righting moment about the assumed axis is obtained.
4. The righting moment divided by displacement gives the righting lever about the assumed axis.

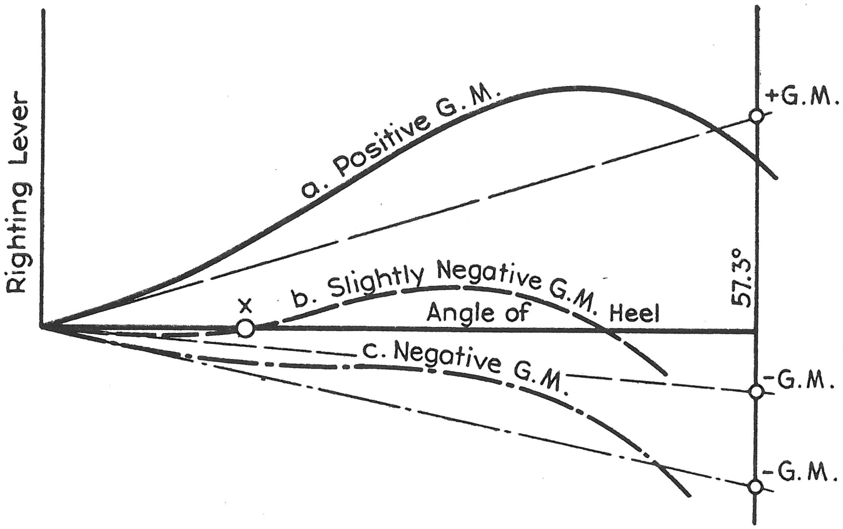


Fig. 13

It will be seen from Fig. 12 that, if the actual VCG is higher than the assumed axis by a distance d, the actual GZ will be less than that obtained using the assumed axis by an amount ($d \sin \theta$), and vice versa. The assumed axis is some-

times taken at the base line so that this correction will always be subtractive.

5. Any watertight erection, as a forecastle, bridge or poop, that is immersed at large angles must be calculated as its length times an average immersed area and added to the molded form.

6. A cross curve is then drawn for each angle of inclination by plotting the values of *GZ* against the corresponding displacement (Fig. 14).

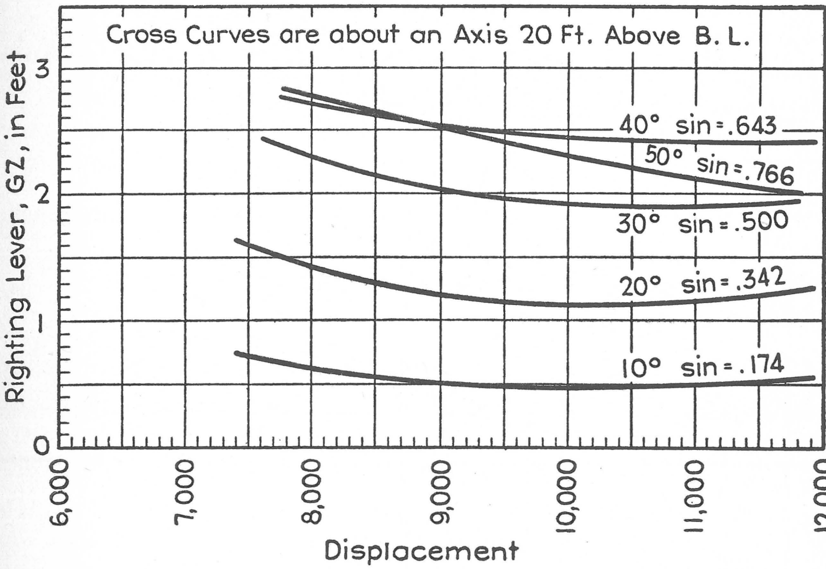


Fig. 14

7. To draw a curve of righting levers for a given condition of loading, a line is drawn across all of the cross curves at the given displacement, the righting levers at each angle read from the cross curves, corrected for the distance between the assumed axis and the actual *VCG* by subtracting (actual *VCG* minus height of axis) times sin θ, and plotted against angle of inclination, as in Fig. 13.

8. To determine the beginning of the curve more accurately, the value of *GM* is laid off at 57.3 degrees, and a line drawn from this to the zero of the diagram as shown.

The curve of righting levers must be tangent to this straight line as shown. For proof, see Reference (*a*), page 120.

If GM is negative, the curve of GZ may look like either curve *b* or curve *c* in Fig. 13. If the negative GM is small, the curve of GZ may cross zero as at x. In such a case the ship, if heeled slowly enough, would not capsize but would heel to the angle where GZ is zero. If GZ is negative at all angles, as in curve C, the ship would capsize.

Period of Rolling

The period of rolling of a ship is given by the formula

$$T = 1.108K \div \sqrt{GM}$$

where

T = period of complete roll, from one side to the other and back, in seconds.
K = transverse radius of gyration of ship, in feet.
GM = metacentric height.

The radius of gyration is the radius of a thin cylinder such that, if the entire ship's weight were concentrated at the cylinder, the moment of inertia of the weight about its CG would be unchanged (see Fig. 15). If the weight were so concentrated, its moment of inertia would be $W \times K^2$. For this moment of inertia to equal its actual value I, K must be $\sqrt{I/W}$.

The radius of gyration is seldom calculated for actual ships. It varies with each condition of loading. It can be inferred approximately from the observed period of roll when GM is known by rearranging the above formula as

$$K = \frac{T \times \sqrt{GM}}{1.108}$$

(Approximately, because the period of actual rolling in water is not quite the theoretical period of unresisted rolling which is given by the formula.)

From such observations, K has been found usually to be

between $0.35B$ and $0.40B$, with $0.38B$ as an average (B = beam of ship). Using $0.38B$ makes $T = 0.42B \div \sqrt{GM}$.

Fig. 16 shows the period of roll associated with a given beam and GM, assuming $K = 0.38B$.

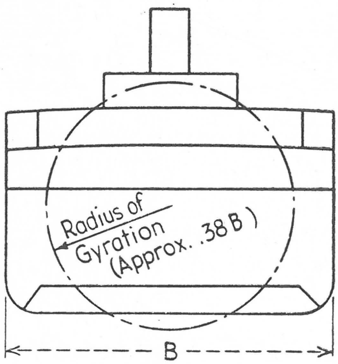


Fig. 15

A short quick period is uncomfortable for passengers, causes racking stresses in the ship and is to be avoided. However, the period is bound to quicken as the size of the ship decreases. For instance, we can easily get a period T of 18 seconds (an easy, comfortable roll) for a 600-foot by 78-foot ship by giving her a GM of about 3.25 feet, while to give a 200-foot by 35-foot vessel the same period would require the dangerously small GM of only 8 inches, and we would be obliged to accept a quicker roll for the smaller vessel.

Good practice is to use a GM in design condition of about $B/20$, which gives a period of roll equal to about $1.88\sqrt{B}$, as is shown in Fig. 16.

Problems

1. Assume that the example ship is loaded to a displacement of 10,000 tons, with a VCG of 18.9 feet corrected for free surface.

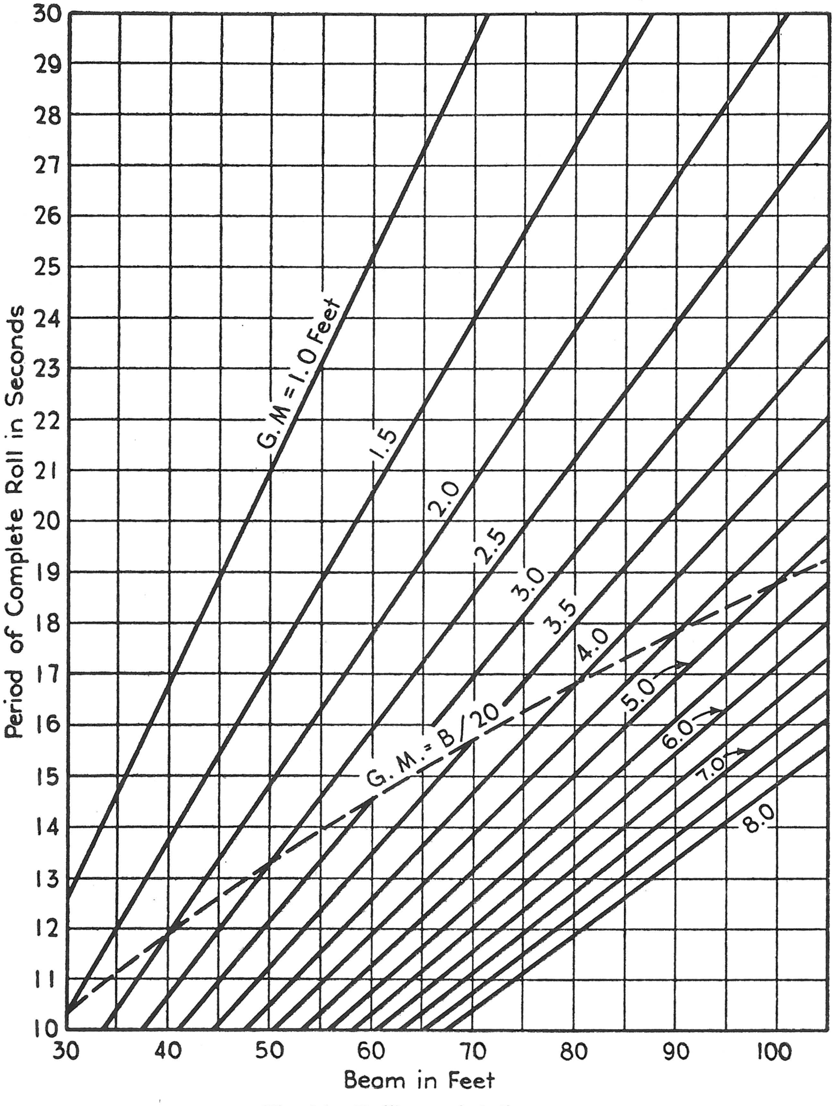


Fig. 16.—Rolling-period diagram

$$T = 1.108K/\sqrt{GM}$$

where T = time of complete (double) roll in seconds
K = radius of gyration in feet
GM = metacentric height in feet
For this diagram K is taken as $0.38B$, so that $T = 0.42B/\sqrt{GM}$.

(a) What is her *GM* in this condition? (Use the curves of form drawn in Chapter II.)

(b) What angle of list will be produced by an off-center weight of 200 tons, 10 feet from the centerline?

2. A vertical-sided double-bottom tank 40 feet wide and 60 feet long is partly filled with fuel oil whose density is 38 cubic feet to the ton.

What is the correction for this free surface on the *GM* of the ship in Problem 1?

3. Assume that the cross curves of the example ship are shown in Fig. 14. Draw a curve of righting levers for the condition given in Problem 1, including tangent at origin. Note that in this case the actual *VCG* is *below* the assumed axis, and the correction is to be *added*. The sine of each angle is noted on the curves for convenience.

4. The example ship is being drydocked at a displacement of 10,600 tons and a *VCG* of 20.2 feet. Assume the ship to settle on the keel blocks simultaneously all fore and aft. (This is not done in practice, but is assumed for simplicity.) What is the *GM* when the water has been pumped down 1 foot, and 3 feet, below the level at which the keel touches the blocks?

Starter: Find the loss in displacement as 12 inches and 36 inches, respectively, times the mean tons per inch for the first foot and first three feet reduction in draft. This loss in displacement equals the upward force on the keel from the blocks. As far as stability is concerned, this is equivalent in effect to removing weight *at the keel.* Subtract this weight at zero height from 10,600 tons at 20.2 feet height, and get a virtual center of gravity of the remaining displacement. Compare with *KM* at the reduced drafts to get *GM.*

This problem will show why supporting blocks are hauled into place under the bilges very soon after the keel has landed on the keel blocks.

Chapter IV

VOLUME CALCULATIONS, TANK AND HOLD CAPACITIES AND TONNAGE CALCULATIONS

References

(*a*) "Principles of Naval Architecture," Volume I, Chapter II.

(*b*) "Measurement of Vessels," pamphlet published by United States Department of Commerce, Bureau of Marine Inspection and Navigation.

The volumes of parts of a ship must be calculated for three purposes:

(a) Volumes of tanks for capacities of fuel oil and water.

(b) Volumes of holds for cargo-carrying capacity.

(c) Entire internal volume of ship (with certain exemptions) for tonnage measurement.

Tank Capacities

These are simply the molded volumes of the spaces concerned, minus a deduction for "internals"; that is, structure and piping within the tank. These capacities may be obtained either by dividing the tank into a suitable number of stations, locating and drawing these stations on the body plan, planimetering them, and applying Simpson's rule to the areas, or by using the regular body-plan stations, and plotting a curve of transverse area of a tank somewhat longer than the tank itself. The area under this curve cut off by the actual tank ends may be in turn planimetered to get the volume of the tank, or the areas at any desired stations can be read from this curve and put through Simpson's rule. In preliminary work, where the extent of the tanks may be changed repeatedly, such a curve of areas is useful in that the effect of moving the tank ends may be quickly obtained.

The gross or molded volume of the tank must be decreased by from 0.5 percent for large tanks to 2 percent for small tanks for "internals," and in the case of fuel oil must be further decreased by about 3 percent (5 percent in naval work) to provide for expansion of the oil when heated.

Capacity Curves. In the case of tanks, it is usual to construct curves showing the capacity corresponding to any depth of liquid. This curve is made by calculating the volume of the tank up to a sufficient number of levels, plotting these volumes against a vertical scale of depth, and drawing a curve through the calculated points. This curve will not necessarily be smooth and fair, for its slope depends on the area of the horizontal planes through the tank. Fig. 17 illustrates the relation between the area of the horizontal planes of a tank and the shape of the capacity curve.

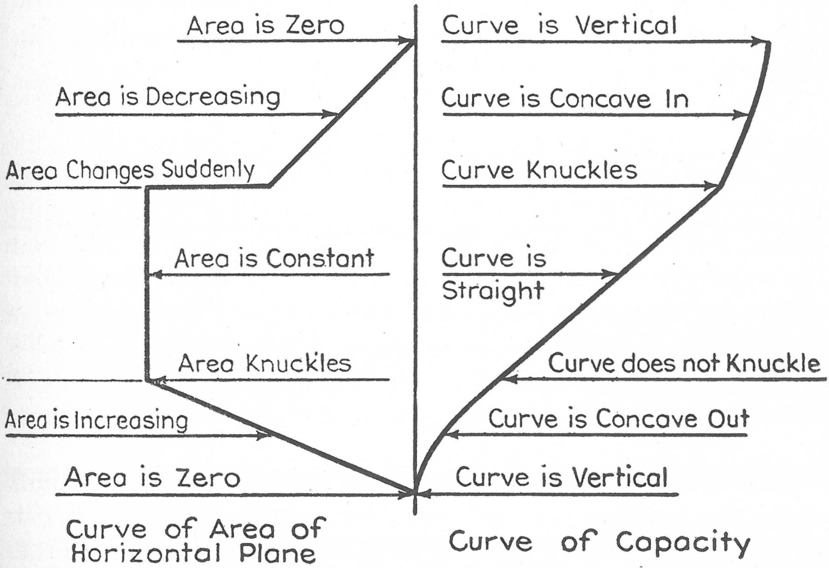


Fig. 17

In practice the curve of area of the horizontal planes is not drawn. It is shown here only to illustrate how the shape of a capacity curve is affected by the shape of the

tank. All of the conditions shown will seldom be found in any one tank.

Sounding Tables. If a tank is fitted with a sounding tube, in which a tape or rod can be lowered to measure the depth of liquid ("sound" the tank), a table is usually prepared giving the capacity of the tank corresponding to every inch (or as desired) of sounding. This is complicated by the fact that the sounding tube is usually sloping, and may have bends in it, so that the sounding tape does not show the vertical depth of liquid. The capacity curve just described must first be drawn. Then the sounding tube is measured on the ship (because it is seldom installed exactly like the drawing) and laid out on the same paper and to the same vertical scale as the capacity curve. Each sloping part of the tube must be swung around into the plane of the drawing so as to show its true slope. Then a scale of feet and inches is laid off on the sounding tube, measuring from the bottom of the tube, and the corresponding capacities read from the capacity curve (see Fig. 18).

Ullage is the distance from a tank top *down* to the top of the liquid in the tank. For oil tankers and barges it is less messy and probably more accurate to measure this distance than to sound the oil, and ullage tables rather than sounding tables are frequently required. These are tables in which the capacity corresponding to a given ullage is tabulated. When the capacity curve has been drawn, the method of making an ullage table is obvious.

Capacities of Holds

Hold capacities are described as "bale" or "grain" capacities. Bale capacities are the volumes inside of the side cargo battens on the frames, above any ceiling on the inner bottom or decks, and below the bottoms of deck beams overhead. Grain capacities are the total molded volumes of holds out to the shell plating and up to the deck plating overhead, with a small deduction for "internals" as in tank capacities.

For bale capacities a special body plan must be drawn showing the inside of battens, etc., and the corresponding transverse areas obtained. From these a curve of trans-

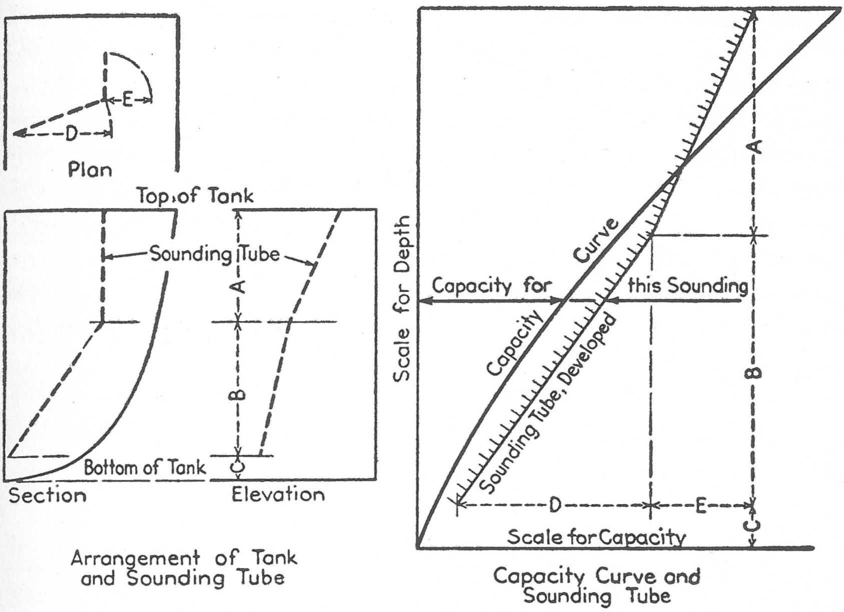


Fig. 18

verse areas of holds is drawn. The extent of the several holds is laid off on this curve, and the corresponding volumes calculated. A deduction must be made for stanchions, pipe casings, etc.

Grain capacities, if required, are obtained in a similar way, except that areas are taken to the shell and decks instead of to the inside of battens and underside of beams.

The capacities of refrigerated holds are obtained similarly, but in this case the capacities are to inside of insulation, coils and coil-protecting battens, so that the capacity becomes only about 60 percent of the molded volume.

For preliminary work it is useful to have coefficients from

previous ships which can be applied to molded volumes to give approximate bale or refrigerated volumes. In the absence of such data, rough type sections can be sketched up, showing frames, beams, brackets, etc., the ratio of the net area to the molded area calculated and used as an approximate coefficient. However, as soon as scantlings have been obtained, the coefficients should be abandoned and actual capacities calculated.

The total hold volume must bear a fairly definite relation to the cargo deadweight. For ordinary general cargo there should be about 60 cubic feet for each ton of cargo, while for special cargoes, such as frozen meat, or chilled fruit, much more space is required.

***CG* of Cargo Capacities.** It is impossible to foretell how cargo will be stowed. In estimating the *CG* of cargo we assume that the stowed cargo will be uniform density. Accordingly, the *CG* of the *cargo spaces* is calculated and taken as the *CG* of the cargo. This is the *CG* of an imaginary homogeneous cargo.

Tonnage

Tonnage (register) is a term applied to the internal volume of a ship, expressed in tons of 100 cubic feet.

Gross tonnage is the entire internal volume, except for certain exempted spaces.

Net tonnage is the tonnage remaining after the nonearning spaces, such as machinery spaces and crew spaces, have been deducted from the gross tonnage. Net tonnage was originally intended as a measure of a ship's earning ability, and is the basis upon which tolls for canal transit, port charges, etc., are based. Accordingly the naval architect tries to design a ship with the least possible net tonnage.

All of the principal maritime governments have their rules describing how tonnage is to be measured. The Suez and the Panama Canal authorities also each have their own rules. By agreement, however, the principal nations may accept each other's measurements. The two canals each require

measurement according to their own rules. Reference (*b*) gives the United States, Suez and Panama rules.

Gross tonnage is obtained in three parts: Underdeck tonnage, 'tween-deck tonnage and enclosed erections. Each is the volume inside of frame battens (if fitted) and the underdeck tonnage is measured above the inner bottom. Each volume is obtained by applying Simpson's rule, in a manner specified in great detail by the rules. Instead of the regular displacement stations, special stations, whose number depends on the ship's length, are used. A special body plan is made, using these stations and showing the lines to which the tonnage is measured.

Exemptions. Certain spaces are not included in gross tonnage. As previously mentioned, the entire double bottom is excluded. Peak tanks fitted only for carrying ballast are excluded. Poops, bridges and forecastles, or, in the extreme case, an entire shelter deck space, if not fitted with watertight enclosures, are not included. (This is not true of the Panama rules.) Certain other spaces as listed in Reference (*b*) are excluded.

Tonnage Openings. As suggested in the preceding paragraph, except for the Panama rules, bridges, etc., or even an entire shelter deck, can be exempted from the gross tonnage (and therefore from the net tonnage) by fitting certain "temporary" and nominally non-watertight openings in the enclosing structure. These openings, consisting either of an un-gasketed bolted plate or of unbattened planks, are described in detail in Reference (*b*).

If such tonnage openings are fitted, the required freeboard *may* be increased (see Chapter XI). This would decrease the draft, and, with it, the weight of cargo that can be carried. In any particular case it is a problem in final cost of operation to decide whether tonnage openings will be beneficial. In general it will depend on the nature of expected cargo. A light cargo, requiring large hold space but relatively light draft, will call for a shelter deck construction with tonnage openings, while a heavy cargo will require

all the attainable draft even at the expense of a larger net tonnage.

Deductions. The spaces whose volume is to be deducted from gross tonnage to get net tonnage are described in detail in Reference (*b*). They consist generally of working spaces and crew's spaces.

The principal deduction is the machinery space.

Instead of deducting fuel spaces, which are also considered to be non-earning, a larger deduction than the actual machinery space is allowed. If the machinery space proper is 13 percent or under of the gross tonnage, the deduction for machinery spaces is $1\frac{3}{4}$ times the actual volume. But if the machinery space volume is over 13 percent, and under 20 percent, of the gross tonnage, the deduction is 32 percent of the gross tonnage. If the machinery space is 20 percent, or over, of the gross tonnage, the deduction may (and does) go back to $1\frac{3}{4}$ times the actual volume. It follows that, if the machinery space is slightly under 13 percent of the gross tonnage, it will pay the owners to increase it to over 13 percent. If possible, in such cases, the required volume of machinery space is made up, not by encroaching on the cargo space, but by including with the machinery space enough of the "light and air" spaces in the boiler and engine casings (which would ordinarily be excluded from all tonnage, both gross and net) to make the required volume. This space must then of course be included in the gross tonnage. Permission must be obtained in each case to do this, and the required procedure is described in Reference (*b*).

Approximate Tonnage

For preliminary work, the estimated tonnage of a proposed ship will be proportioned from the actual tonnage of a similar ship, and the naval architect will have accumulated data for this purpose.

The (under-deck plus 'tween-deck) tonnage may be proportioned to ($L \times B \times D \times$ block coefficient). The erec-

tions will be individually estimated. The "propelling-power" or machinery-space deduction will ordinarily be assumed to be 32 percent of the gross tonnage. The crew space and working space will be estimated from the type ship in proportion to $(L \times B \times D)$.

If no similar ship data are available, the tonnage must be estimated by drawing tonnage sections (to the inside of side battens, top of inner-bottom ceiling, and top of beams) on the proposed lines. In this case no attention need be paid to the special stations required by the measurement rules; the regular displacement stations can be used. The areas may be obtained by planimeter and the volumes by Simpson's rule. The erections will be estimated as the product of the inside length, width (average) and height. The machinery space will be checked to see that it is at least 13 percent of the gross tonnage, and 32 percent used for a deduction. Crew and working spaces will be estimated from the list in the rules as well as possible.

Data from similar ships are extremely desirable for estimating approximate tonnage of proposed ships.

Problems

1. Draw a capacity curve and construct a sounding-tube table with intervals of 6 inches for the tank shown in Fig. 19. Express capacities as tons of fuel oil at 38 cubic feet per ton. Deduct 1 per cent for internals. See suggestions below.

2. Assume that the tank is filled just to the knuckle in the tank. Where is the transverse virtual center of gravity (or transverse metacenter) of the liquid?

3. Assume that the tonnage data for the example ship are as shown. Approximate the gross and net tonnage for a similar, but larger, ship of the dimensions shown. See suggestions below.

	Example Ship	Proposed Ship
LBP × *B* × *D*	420′ × 54′ × 30′ 3″	440′ × 55′ × 34′
Block coefficient	0.762	0.73
Under and 'tween deck tonnage	5100	
Poop, bridge and forecastle	60	
Houses and excess hatches	110	
Light and air included for 32 percent deduction	80	
Gross tonnage	5350	
Propelling-power deduction	1710	
Crew spaces	150	
Working spaces	50	
Total deduction	1910	
Net tonnage (gross tonnage minus total deductions)	3440	

Suggestions for Problem 1: The capacity curve will be a straight line from the knuckle in the tank to the top, and no intermediate levels need be calculated. One intermediate level between the knuckle and the bottom should be sufficient, provided the principles shown in Fig. 17 are kept in mind.

Suggestions for Problem 3: Since the proposed ship is assumed to be similar to the example ship, the "poop, bridge and forecastle" tonnage, and that of the "houses and excess hatches" may be taken as varying with $L \times B$, for

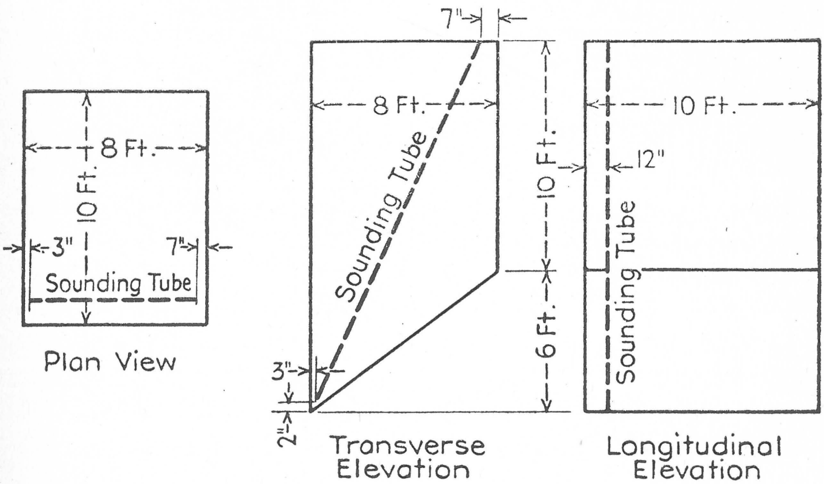


Fig. 19

the 'tween-deck height would be the same in both cases. The "light and air" volume may be proportioned to the "under and 'tween deck tonnage."

(In practice, as soon as the design is sufficiently definite, these preliminary figures would be checked against the actual arrangement and revised as necessary.)

Chapter V

LOCAL STRENGTH AND AMERICAN BUREAU OF SHIPPING RULES

REFERENCES

(*a*) American Bureau of Shipping "Rules for Building and Classing Steel Vessels."
(*b*) Timoshenko, "Strength of Materials" (2 volumes).
(*c*) Carnegie "Pocket Companion."
(*d*) Rossell, "Riveting and Arc Welding."
(*e*) "Specifications, for Riveting Vessels of U. S. Navy" (General Specifications, Appendix 4).*
(*f*) "Specifications for Welding Vessels of U. S. Navy" (General Specifications, Appendix 5).*
(*g*) Lovett, "Applied Naval Architecture," Chapter XI.
(*h*) Murray, "Strength of Ships."

The subject of strength of ships may be divided into two broad divisions: Local strength of a part to carry its load, as, for example, a deck beam or a pillar; and the strength of the "ship girder" to withstand stresses caused by large waves at sea.

In practice, a weight calculation (see Chapter VI) must precede calculation of the strength of the ship girder, and, before weights can be estimated, scantlings of individual members must be determined. This chapter, therefore, considers local strength, and the hull girder calculation is discussed in Chapter VII.

In ordinary merchant ship design, most scantlings are determined not by strength calculations but by the rules of a classification society.

Each principal maritime nation has such rules. In this country the American Bureau of Shipping (A.B.S.) rules

* **These are not "confidential."**

are usually used, although occasionally compliance with Lloyd's rules (British) is specified. We will use the American Bureau of Shipping rules, Reference (*a*). These rules enable us to dispense with calculations of stresses, not only in beams, stanchions, frames, etc., but also in the hull girder, for they specify the scantlings of shell plating, strength decks and other structures which constitute the hull girder.

However, the rules go only to a length of 750 feet, and larger ships must be designed from a strength calculation. Also, frequently some member must be designed without the aid of the rules. So the underlying principles, known as "Strength of Materials," must be understood by the naval architect.

The subject of "Strength of Materials" fills entire books [see Reference (*b*)]. We will go only far enough to be able to design a simple beam or column, proportion rivets and welding, and understand the standard hull girder calculation.

We must distinguish between force, stress and strain. Force is the total load coming on a member. Stress is the force per unit area, as pounds per square inch, and strain is the change in length per unit length resulting from the stress. For many materials, including steel, stress and strain are proportional up to a stress called the proportional limit. The ratio of stress to strain is called the modulus of elasticity, denoted by E, and sometimes called Young's modulus. It can be thought of as the stress which would double the length of a specimen, if stress and strain remained proportional up to such a stress. For steel, E is about 29,000,000 pounds per square inch.

There are three kinds of stress: tension, compression and shear, all defined in Chapter I. Bending usually involves all three (see Fig. 20).

The bottom fibers of the beam shown in Fig. 20 are obviously in tension, the top fibers are in compression, and the entire beam is subject to both vertical shear and horizontal shear.

As the top fibers are in compression, and the bottom fibers

in tension, there must be fiber between which is not stressed at all. This unstressed fiber is called the neutral axis, and for a straight beam can be proved [see Reference (*b*)] to be at the center of gravity of the section.

Fig. 21(e) shows a beam freely supported at its ends and carrying a single concentrated load. Equilibrium requires that the sum of the reactions R_1 and R_2 equal the load W, and that the moment of either reaction about the opposite end of the beam be equal and opposite to the moment of the load W about the end; that is,

$$R_1 \times l = W \times b$$

and

$$R_1 = W \times b/l$$

Similarly,

$$R_2 = W \times a/l$$

If there are several loads, the reactions are obtained in the same way by dividing the sum of the moments of all the loads about one end by the length of the beam to get the reaction at the other end.

Vertical Shear and Shear Diagram

It can be seen from Fig. 20(c) and from the definition of equilibrium that at any section in a beam there must be a vertical shearing force which is equal and opposite to the reaction at either end minus any loads that occur between that end and the section in question. A shear diagram can be drawn on a base line representing the length of the beam whose ordinate at any point represents the vertical shear at that point. Fig. 21 shows shear diagrams for each kind of loading shown.

Bending Moment and Bending Moment Diagram

It can also be seen from Fig. 20(a) and from the definition of equilibrium that at any section in a beam there must be

a bending moment equal to the moment of the reaction at either end about the section in question minus the moment of any loads that occur between that end and the section. A bending moment diagram can be drawn

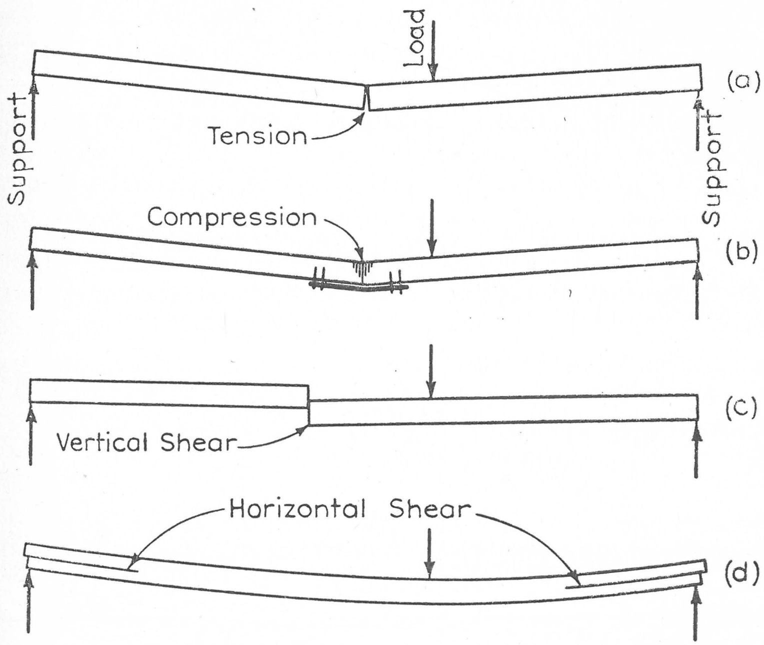


Fig. 20

whose ordinate at any point represents the bending moment at that point. Fig. 21 shows bending moment diagrams for each kind of loading shown.

The bending moment at any section must be resisted by an equal moment of the tension and compression within the beam. We now assume that *stress in the beam is proportional to distance from neutral axis*. It can be proved [see Reference (*a*)] to follow from that assumption that the

$$\text{Maximum stress } p = \frac{My}{I}$$

This is an important equation. I and y depend only on the shape of the section, and are often combined into the

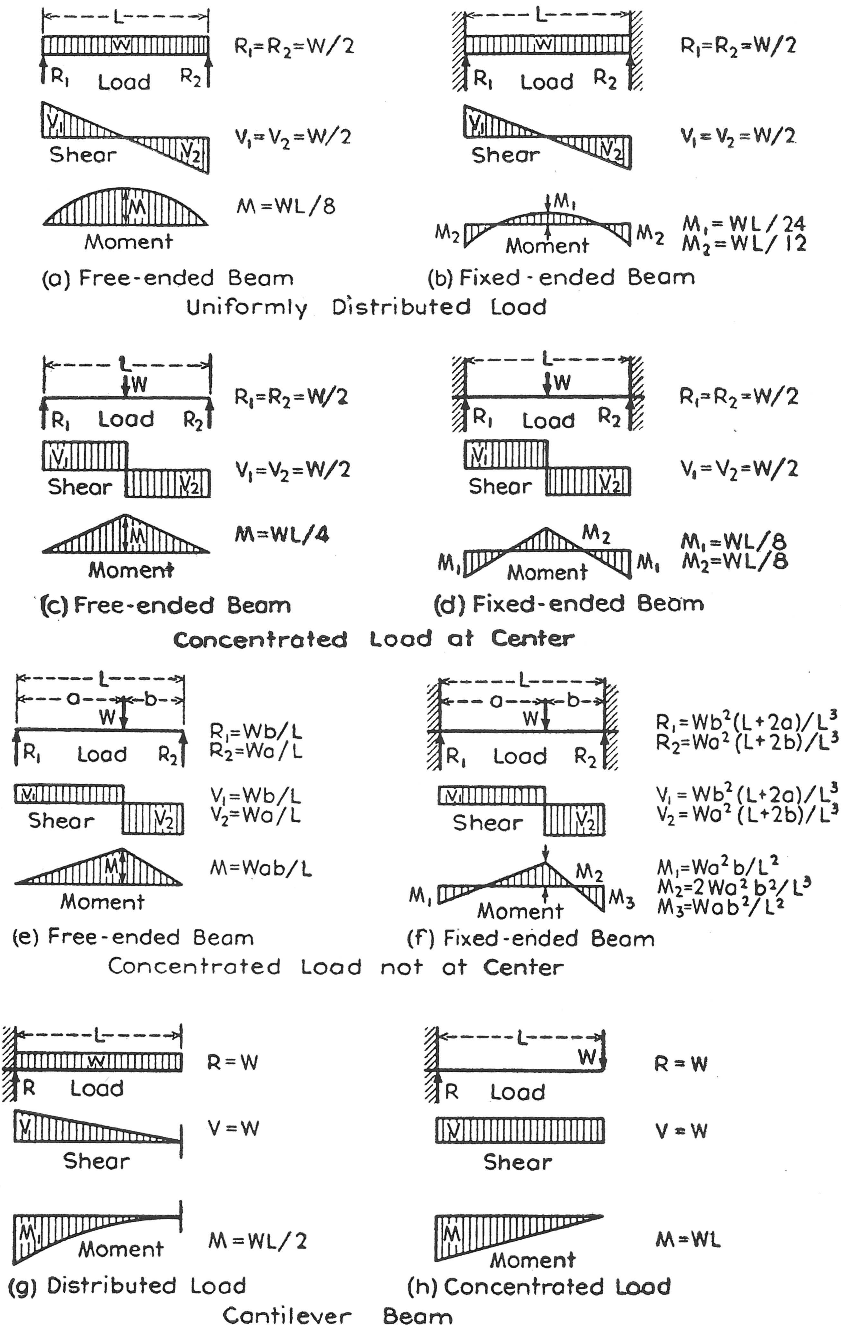

L
W
R1=R2=W/2
R1 Load R2
V1 V2
Shear
V1=V2=W/2
M
Moment
M=WL/8
M1 M2
M1=WL/24
M2=WL/12
(a) Free-ended Beam
(b) Fixed-ended Beam
Uniformly Distributed Load
M=WL/4
M1=WL/8
M2=WL/8
(c) Free-ended Beam
(d) Fixed-ended Beam
Concentrated Load at Center
a b
R1=Wb/L
R2=Wa/L
V1=Wb/L
V2=Wa/L
M=Wab/L
R1=Wb²(L+2a)/L³
R2=Wa²(L+2b)/L³
V1=Wb²(L+2a)/L³
V2=Wa²(L+2b)/L³
M1=Wa²b/L²
M2=2Wa²b²/L³
M3=Wab²/L²
(e) Free-ended Beam
(f) Fixed-ended Beam
Concentrated Load not at Center
R Load
R=W
V=W
M=WL/2
M=WL
(g) Distributed Load
(h) Concentrated Load
Cantilever Beam

Fig. 21

section modulus, S, equal to I/y. The section modulus for all rolled sections, such as I-beams, angle bars, etc., is tabulated in handbooks. To select a beam, the maximum bending moment is divided by the desired stress, giving the required section modulus, and a beam having that section modulus selected from the available sizes.

The maximum bending moment is obtained from a bending moment diagram such as shown in Fig. 21. Similar diagrams for a variety of loadings are published in reference books such as Reference (*c*). Sometimes, as in the hull girder calculation in Chapter VIII, the maximum bending moment is found from an interesting and useful relation between the shear diagram and the bending moment. In Fig. 21(e), for example, the area under the left-hand portion of the shear diagram is $(Wb/l) \times a$, or Wab/l, and this is also the value of the maximum bending moment. This can be proved to be generally true; that is, *the bending moment at any point in a free-ended beam is numerically equal to the area under the shear curve up to that point.* In many cases this fact furnishes the easiest method of obtaining the bending moment diagram. The maximum moment must occur at a point where the shear curve crosses zero.

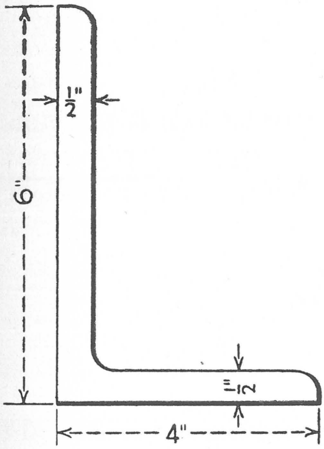


Fig. 22

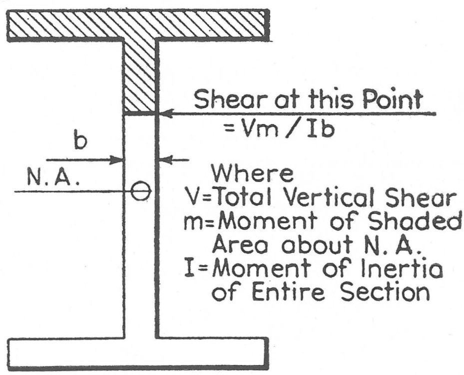


Fig. 23

To calculate I for built-up sections, as a plate girder, or, as an extreme case, the hull girder, the section is treated as a combination of rectangles, the moment of inertia of each of which, about the center of gravity of the entire section, is $(bd^3/12) + Ax^2$, where b is the breadth, d the height, A the area, and x the distance from the center of gravity of the individual rectangle to the combined center of gravity or neutral axis of the entire section.

For example, the moment of inertia of a 6-inch × 4-inch × ½-inch angle (Fig. 22), given in Reference (a) as 17.4, would be found as follows:

	Size	Area	Arm	Moment	$\frac{bd^3}{12}$	x	Ax^2
Vertical flange	6 × ½	3.00	3.00	9.00	9.00	1.01	3.06
Horizontal flange	3½ × ½	1.75	0.25	0.44	0.03	1.74	5.30
		4.75	1.99	9.44	9.03		8.36
						9.03 + 8.36 =	17.39

One further point about shear must be brought out [see Fig. 20(c and d)]. It can be shown that shear is not uniformly distributed over a section, and that at any point within a beam the vertical shear and horizontal shear are equal, and (see Fig. 23)

$$\text{sheer stress } q = \frac{Vm}{Ib}$$

where

V = total vertical shear.
m = moment of shaded area outside of point in question about the center of gravity of the section.
I = moment of inertia of entire section.
b = breadth of section at point in question.

In flanged sections such as channels and I-beams, this formula will be found to make the web carry practically all of the shear and for approximate work such webs are assumed to carry the entire shear uniformly distributed.

Working Stresses

The ultimate or breaking strength of ordinary mild steel is about 60,000 pounds per square inch, and the elastic limit and the yield point (see definitions) are both slightly over 30,000 pounds per square inch. Both values can be greatly increased by special treatment: Bridge cable can be given an ultimate strength of 200,000 pounds per square inch and a yield point of 100,000 pounds per square inch. The usual working stress is about one-fourth the ultimate stress or one-half the yield point stress; that is, about 15,000 pounds per square inch for mild steel. This gives a factor of safety of about 4, if based on the ultimate strength, or about 2, if based on the yield point.

Strength of other materials is tabulated in reference books, such as Reference (*c*). An average working stress for many kinds of wood is about 1000 pounds per square inch.

Columns. A column is a relatively slender compression member. A common wooden yardstick, whose cross section is about 1 inch by ⅛ inch, is an illustration. In tension such a stick can carry 200 pounds easily, but in compression it will not carry the pressure that can be exerted by one's hand. The idea of "critical load" can be well illustrated with such a yardstick. Stand one erect on the floor and press on the upper end, gradually increasing the pressure. The stick will return the pressure without bending until a certain definite pressure is reached, when it will start to bend instead of resist. Two basic facts will be noticed:

(*a*) This critical load depends only on the stiffness of the stick and has no relation to its ultimate strength.

(*b*) The least stiffness (the thin way) controls.

A measure of slenderness is required, which should be the ratio of length to some transverse dimension involving I. For this the radius of gyration is used, which is the distance from the center of gravity to the point where all the area might be concentrated without changing I. This definition makes the radius of gyration, r, equal to $\sqrt{I/A}$.

The yardstick is an extreme example; ordinary columns

are much less slender, and carry their load by a combination of compression and column action. l/r is called the slenderness ratio, and in practice is never over 200, and rarely over 120. Such columns are designed by empirical formulas; that is, formulas constructed so as to agree with the results of tests. The formula adopted for allowable stress by the American Institute of Steel Construction [Reference (*c*), page 281] is

$$p = \frac{18{,}000}{1 + \frac{1}{18{,}000}\left(\frac{l}{r}\right)^2}$$

which gives 15,000 pounds per square inch allowable stress when $l/r = 60$; 10,000 pounds at $l/r = 120$; and 5600 pounds at $l/r = 200$.

This kind of formula requires a trial-and-error method of design: a trial column section is selected, its l/r and the resulting permissible stress obtained and compared with the actual stress obtained by dividing the load by the area of the column. If the actual stress is the higher of the two, a larger section is tried.

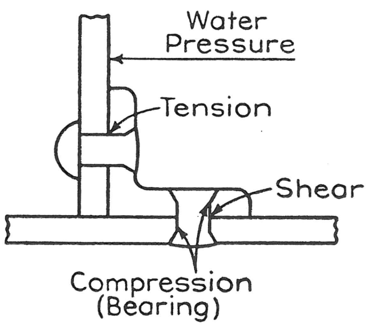


Fig. 24

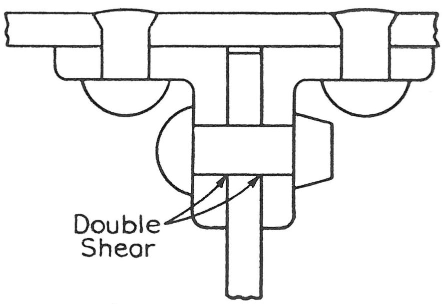


Fig. 25

Riveting. A riveted joint may involve all three kinds of stress, as shown in Fig. 24. Rivets are not intentionally subjected to bending stresses.

A riveted seam may fail in four ways:

1. The plate or rivet may crush where they bear on each other.
2. All rivets may shear off.
3. The rivets may tear through the plate.
4. The plate may tear at the outer row of rivets.

If there are several rows of rivets, the joint may fail in a combination of these ways, such as tearing at an inner row and shearing the rivets in an outer row. Each possible method of failure must be investigated. The strength of the weakest, compared to the strength of the solid plate, is the efficiency of the joint.

The first type of failure may be avoided by making the rivets small enough in relation to the thickness of plate so that their strength in shear is not greater than in bearing. Then the second type can be prevented by having enough rivets, using more rows if necessary. The third type can be prevented by making the "edge distance" from the edge of the plate to the center of the rivet at least 1½ times the rivet diameter. This leaves the fourth type of failure as determining the possible efficiency of the joint, which can never be more than $(S - 1)/S$, where S is the spacing in the outer row in diameters.

Maximum efficiency is not needed in ship work, for the plating generally is weakened to about 85 percent efficiency by the rivets, spaced about 7 diameters, attaching it to framing and deck beams. Rivet sizes and spacing, and number of rows, are specified in Reference (*a*) for merchant vessels, and in Reference (*e*) for naval vessels.

In watertight or oiltight work, the spacing is limited, regardless of strength, to that which will prevent bulging of the calked edge between the rivets, this breaking the calking. This spacing is specified in Reference (*a*) to be 4½ diameters for watertight work.

If a joint is weak regarding type 2 failure only, it may be improved by placing the rivets in double shear as in Fig. 25. The rivet then must shear twice simultaneously. The strength in double shear is found experimentally to be only

about 1.8 times the strength in single shear [Reference (*d*), page 17]. Strength against the other types of failure is not improved by using double shear.

Suitable stresses for rivet design are [Reference (*d*), page 17]:

12,000 pounds per square inch in single shear
15,000 pounds per square inch in tension
24,000 pounds per square inch in bearing

Welding. Welding, like rivets, is designed for tension, compression and shear, but not bending. The common weld is the fillet, Fig. 26(a) or (b), and the butt-weld, Fig. 26(c). Design is based on uniform direct stress across the narrowest part of the weld. Actually, the stress distribution is complicated and uncertain, but this is cared for by using stresses obtained experimentally. The ultimate stress varies from 40,000 pounds per square inch for large welds to 55,000 pounds per square inch for small welds, using covered electrodes. (Covered electrodes are coated with a covering which gives off gases which exclude air from the arc and the weld, improving the quality of the weld. Covered electrodes only are used on all important work.) For general design a working stress of 10,000 pounds, regardless of type of stress, is convenient and conservative.

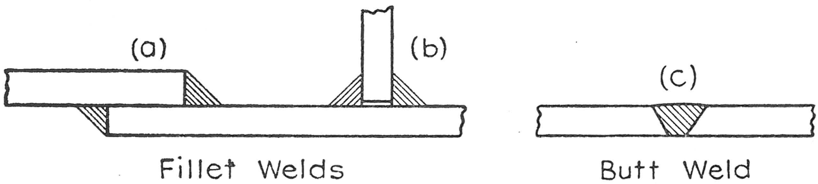


Fig. 26

While the efficiency of a riveted joint must be less than 100 percent, welded joints can easily be made 100 percent efficient.

Sizes of welds are specified in Reference (*a*) for merchant vessels, and in Reference (*f*) for naval vessels.

This is as far as we will go into "Strength of Materials." Many points have not been discussed: beams continuous over several supports, critical stresses in plating where it begins to buckle under shear or compression, etc. We have discussed only what is needed to understand the rest of the book, and the student who wants to go further should get one of the many standard books on the subject.

Problems on Strength of Materials

1. A beam must be selected to carry a uniformly distributed load of 5000 pounds on a span of 16 feet 8 inches, with freely supported ends.

The only beams available are a

6-inch channel with an I of 25.3 inches^4
10-inch channel with an I of 101.8 inches^4
10-inch I-beam with an I of 122.1 inches^4

Which should be chosen, and what is the resulting stress?

Note: From Fig. 21(a), the maximum bending moment in such a beam is $wl/8$.

2. A circular solid steel column 10 feet long is needed to carry a load of 75,000 pounds. What diameter is needed?

Note: The radius of gyration of a circle is one-half the radius of the circle.

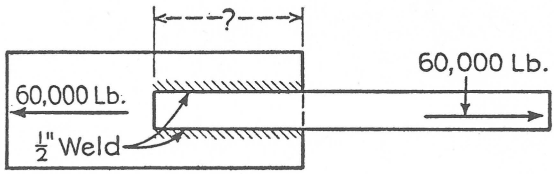


Fig. 27

3. American Bureau of Shipping rules [Reference (*a*)] specify for an end lap in 0.50-inch shell plating, 3 rows of ¾-inch rivets, spaced four diameters (3 inches) center to center. What is the weakest mode of failure and the effi-

ciency? Take the area of a driven ¾-inch rivet as that of a 13/16-inch circle, or 0.519 square inch and the cross-sectional area of a 13/16-inch diameter countersunk hole in 0.50-inch plate as 0.515 square inch. Take the plate strength as 60,000 pounds per square inch and shear of rivets at 48,000 pounds. Consider only shear of all rivets and tearing of plate, as we may assume that the rule edge distance and relative size of rivet are proper.

4. In Fig. 27, assume ½-inch weld along both sides. How long a lap is required?

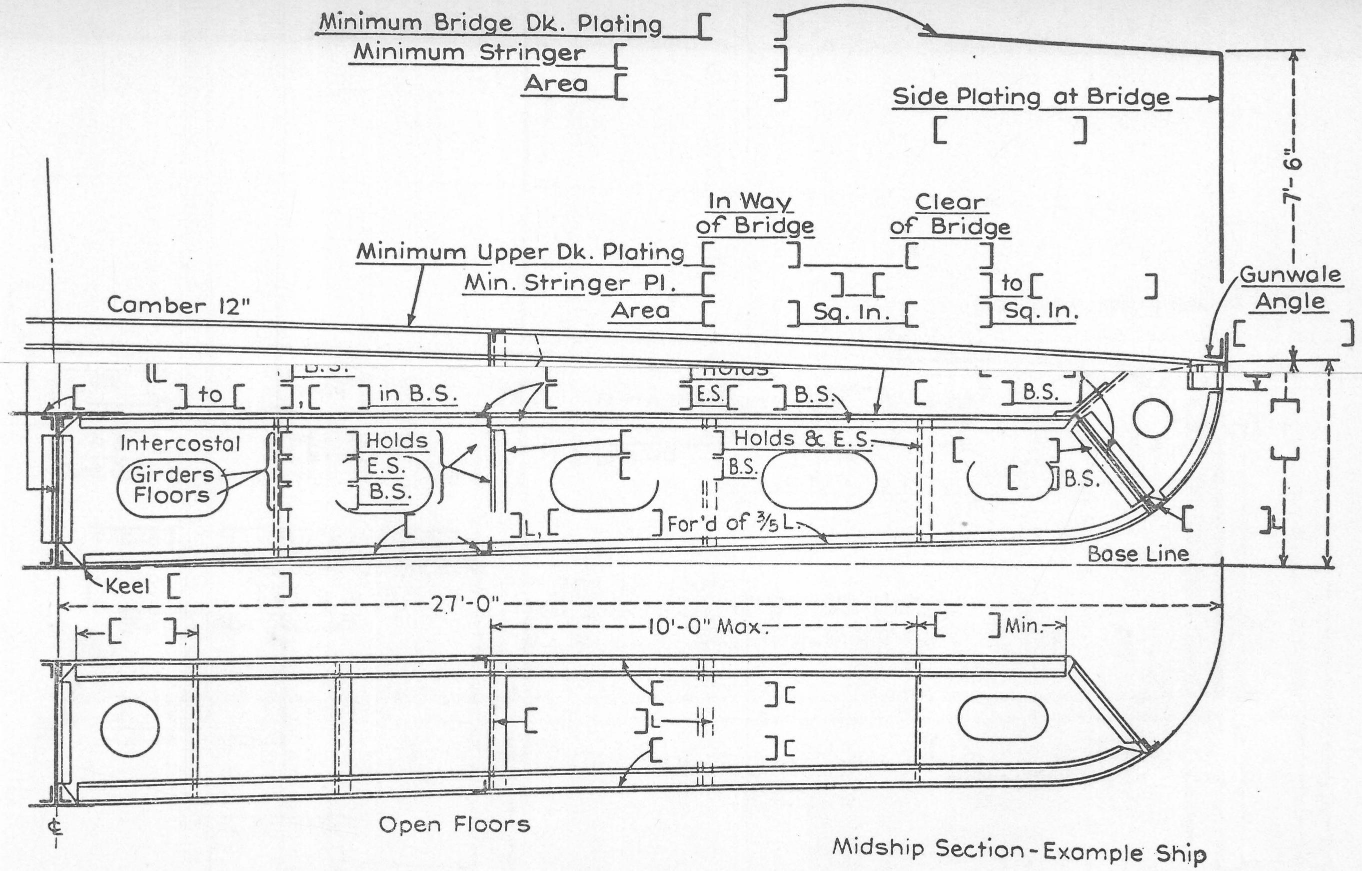
American Bureau of Shipping Rules

The next part of this chapter cannot be worked without a copy of Reference (*a*). These rules are available in every naval architect's office, and can be bought from the American Bureau of Shipping, 47 Beaver Street, New York. The following is based on the 1942 edition. These rules give the required scantlings for all the principal parts of a ship from 100 feet to 750 feet long, as well as rules for riveting and welding, rules for testing materials, and the required sizes of anchors and cables. We will apply these rules, in part, to our example ship.

Midship Section. A midship section of the example ship is shown in Fig. 28. In designing a ship this is the first structural drawing made. On a large complicated ship several sections, and, in addition, scantling plans consisting of deck plans and a profile showing pillars and girders would be drawn. We will place on Fig. 28, from the rules, scantlings of the keel, shell, inner bottom, floors, longitudinals, frames and decks, and on Fig. 29 scantlings for an ordinary watertight bulkhead.

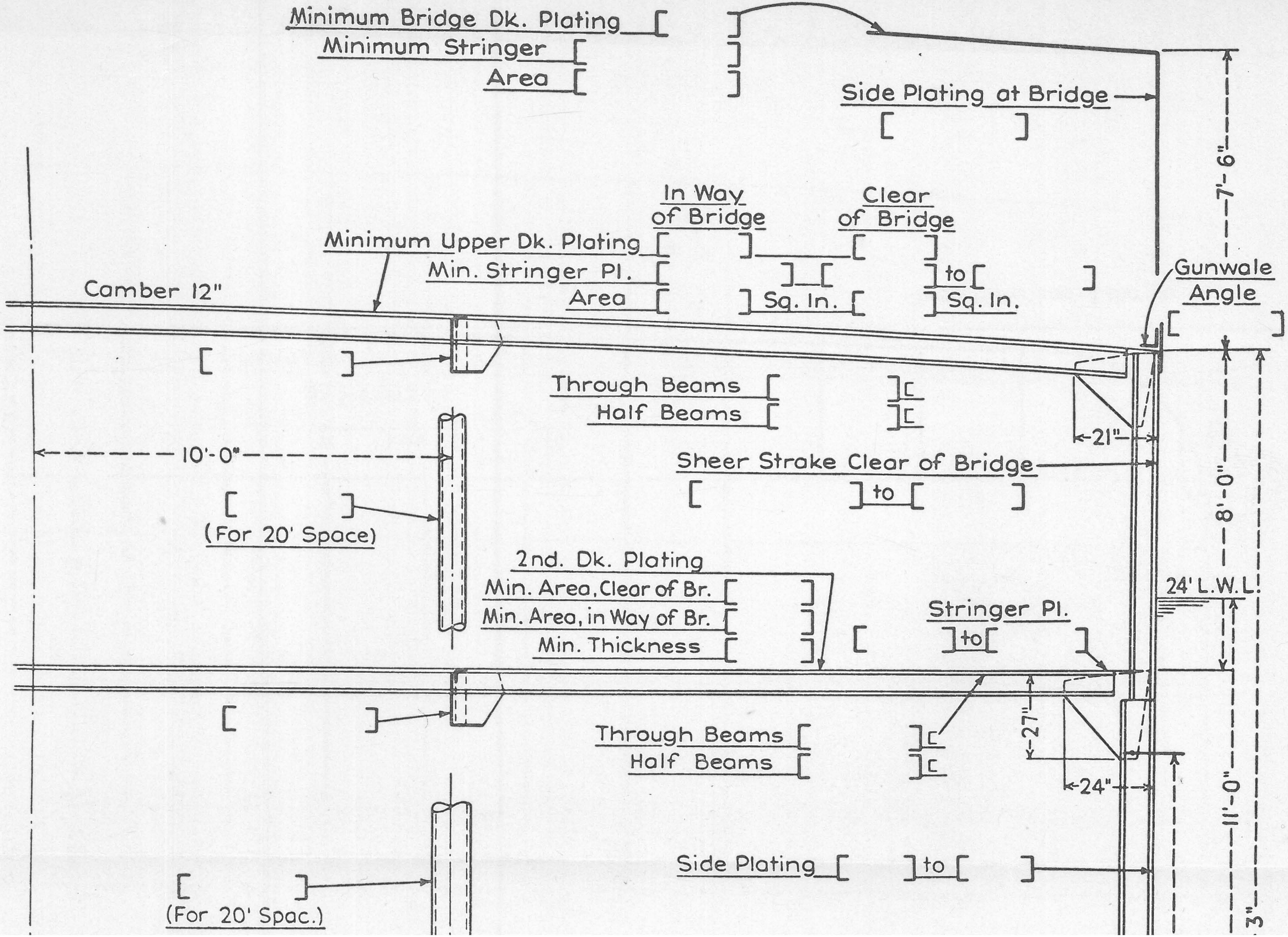
Application of the Rules. Obtain a copy of the American Bureau of Shipping rules, Reference (*a*). Section 2 defines certain dimensions and terms. Our proportions will be found to conform to the requirements of Section 2(10).

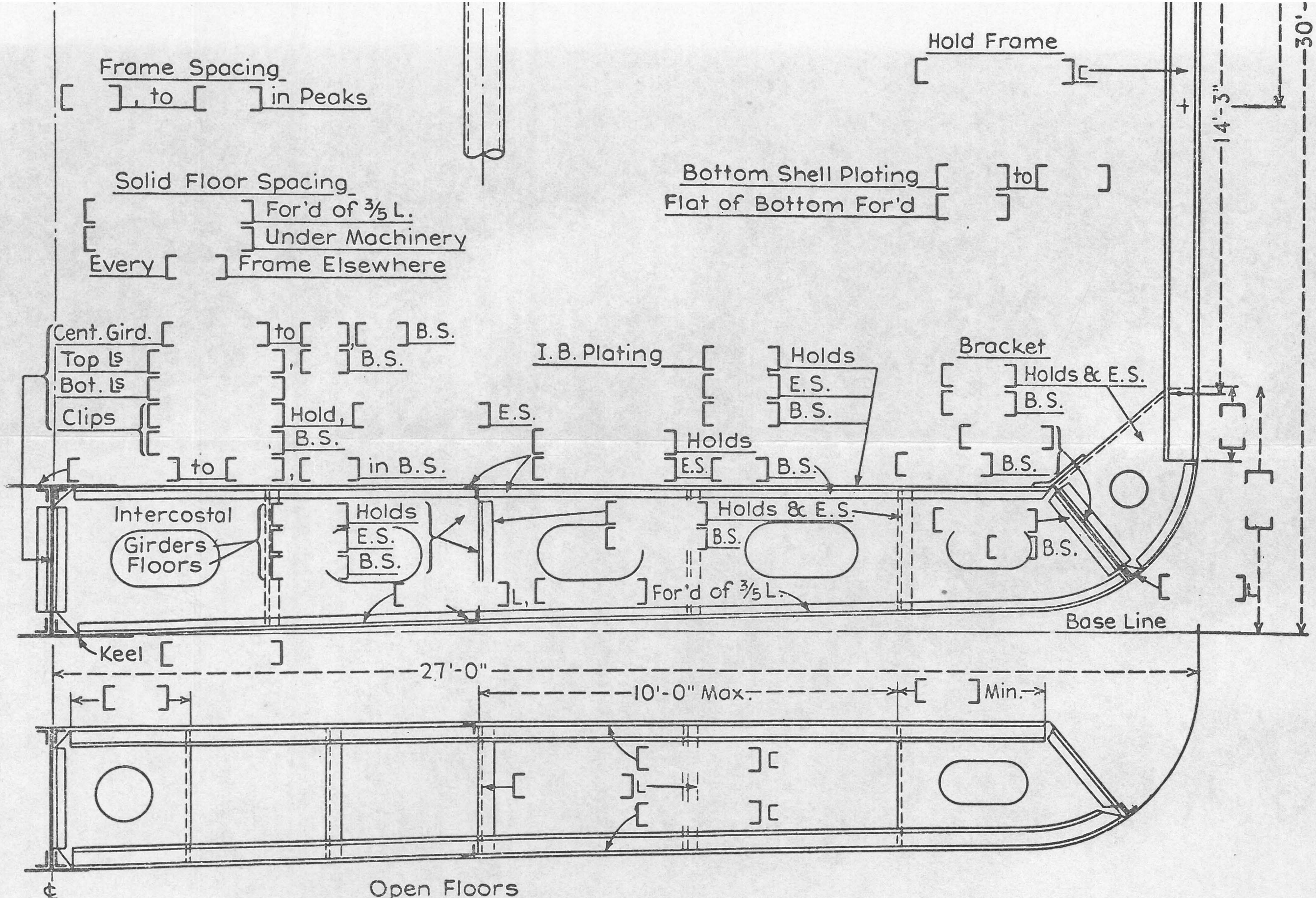
Note particularly Section 3(2). The usual way of stating that a member has a given size throughout the midship half-length and then tapers to a smaller size at the ends is as



Minimum Bridge Dk. Plating
Minimum Stringer
Area
Side Plating at Bridge
7'-6"
In Way of Bridge
Clear of Bridge
Minimum Upper Dk. Plating
Min. Stringer Pl.
Area
Sq. In.
to
Sq. In.
Gunwale Angle
Camber 12"
to
in B.S.
E.S.
B.S.
Intercostal
Girders
Floors
Holds
E.S.
B.S.
Holds & E.S.
B.S.
L,
For'd of 3/5 L.
B.S.
Base Line
Keel
27'-0"
10'-0" Max.
Min.
Open Floors
Midship Section - Example Ship

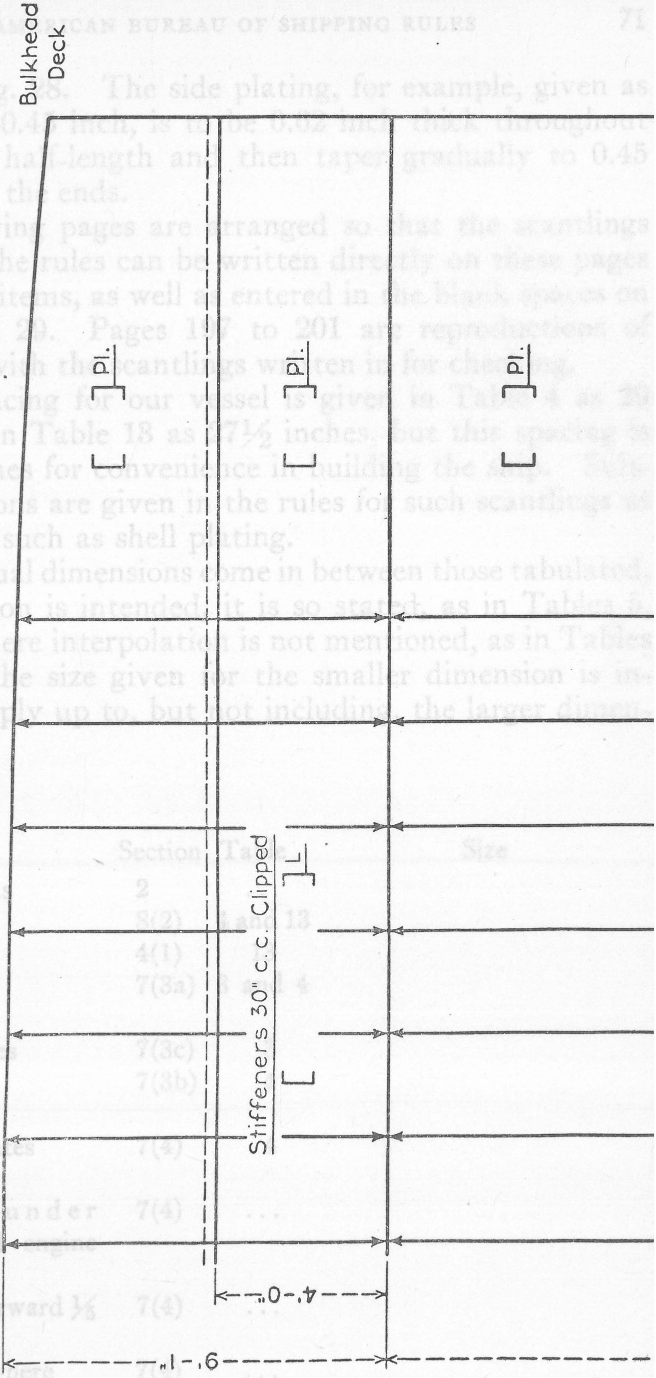
Fig. 28


Minimum Bridge Dk. Plating
Minimum Stringer
Area
Side Plating at Bridge
7'-6"
In Way of Bridge
Clear of Bridge
Minimum Upper Dk. Plating
Min. Stringer Pl.
to
Area
Sq. In.
Sq. In.
Gunwale Angle
Camber 12"
Through Beams
Half Beams
21"
10'-0"
Sheer Strake Clear of Bridge
to
(For 20' Space)
8'-0"
2nd. Dk. Plating
Min. Area, Clear of Br.
Min. Area, in Way of Br.
Min. Thickness
24' L.W.L.
Stringer Pl.
to
27"
Through Beams
Half Beams
24"
11'-0"
Side Plating
to
(For 20' Spac.)
3"



Midship Section - Example Ship

Fig. 28



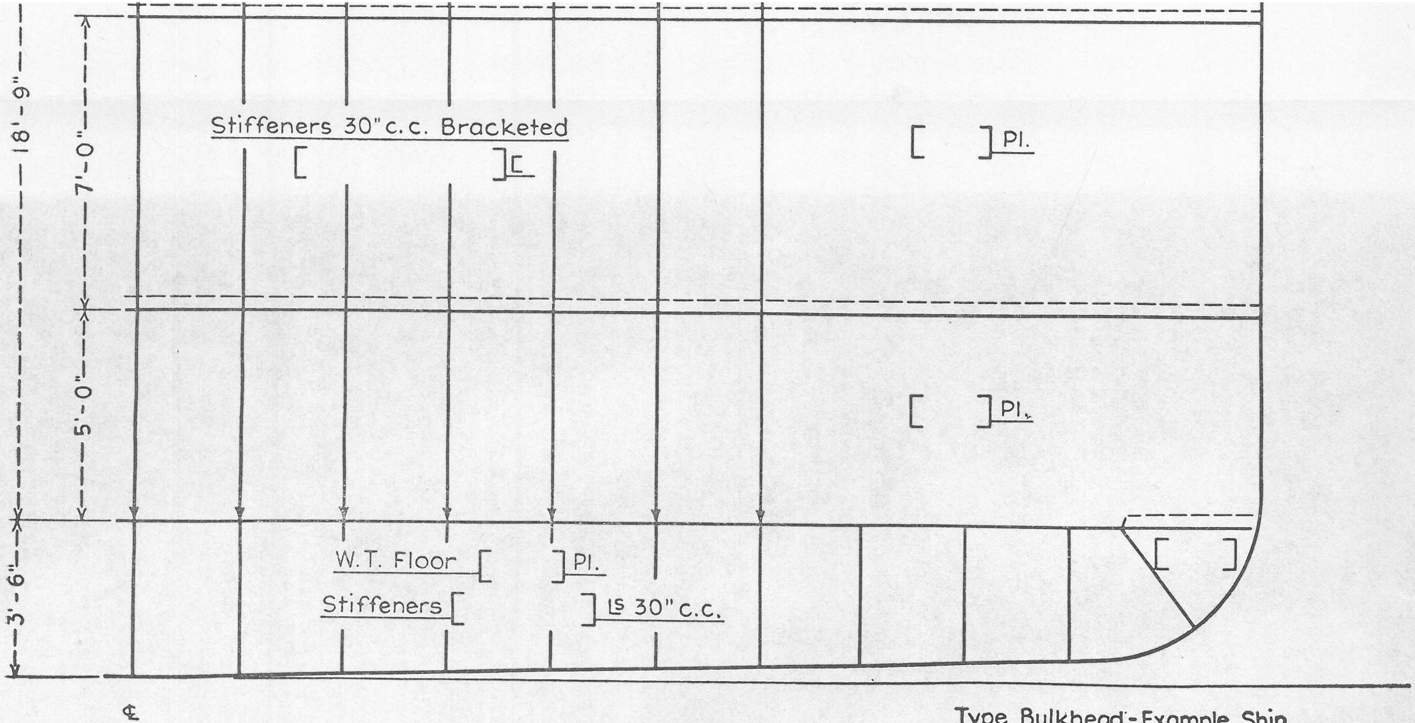
shown on Fig. 28. The side plating, for example, given as 0.64 inch to 0.45 inch, is to be 0.62 inch thick throughout the midship half-length and then taper gradually to 0.45 inch thick at the ends.

The following pages are arranged so that the scantlings taken from the rules can be written directly on these pages opposite the items, as well as entered in the blank spaces on Figs. 28 and 29. Pages 197 to 201 are reproductions of these pages with the scantlings written in for checking.

Frame spacing for our vessel is given in Table 4 as 29 inches, and in Table 18 as 27½ inches but this spacing is made 30 inches for convenience in building the ship. Suitable corrections are given in the rules for such scantlings as are affected, such as shell plating.

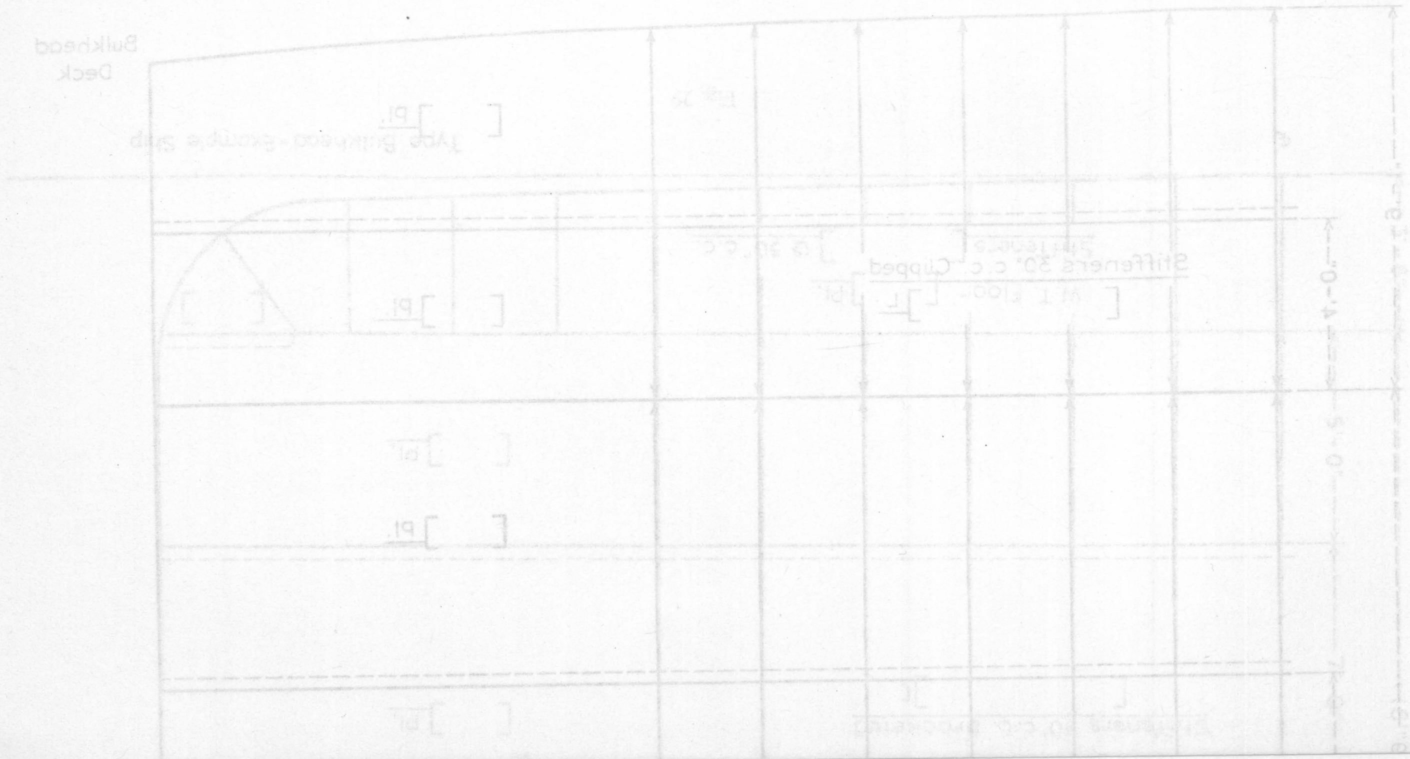
Where actual dimensions come in between those tabulated, if interpolation is intended, it is so stated, as in Tables 5, 12, etc. Where interpolation is not mentioned, as in Tables 1, 16, etc., the size given for the smaller dimension is intended to apply up to, but not including, the larger dimension.

Item	Section	Table	Size
Rule dimensions	2		
Frame spacing	8(2)	4 and 18	
Flat plate keel	4(1)	13	
Center girder	7(3a)	3 and 4	
Bottom angles	7(3c)		
Top angles	7(3b)		
Solid floor—Plates	7(4)		
Spacing—under boiler and engine bearers	7(4)	. . .	
Spacing in forward ⅕ length	7(4)	. . .	
Spacing elsewhere	7(4)	. . .	



Type Bulkhead-Example Ship

Fig. 29



Stiffeners 30" c.c. Clipped
W.T. Floor

shown on Fig. 28. The side plating, for example, given as 0.64 inch to 0.45 inch, is to be 0.62 inch thick throughout the midship half-length and then taper gradually to 0.45 inch thick at the ends.

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Item	Section	Table	Size
Rule dimensions	2	...	
Frame spacing	8(2)	4 and 13	
Flat plate keel	4(1)	13	
Center girder	7(3a)	3 and 4	
Bottom angles	7(3c)	3	
Top angles	7(3b)	4	
Solid floor—Plates	7(4)	4	
Spacing—under boiler and engine bearers	7(4)	...	
Spacing in forward ⅕ length	7(4)	...	
Spacing elsewhere	7(4)	...	

Item	Section	Table	Size
Frame bars	7(4f)	4	
Reverse frames	7(4h)	4	
Stiffeners	7(4d)	4	
Clips to center girder	7(4e)	25K	
Clips to margin	. . .	25K	
Clips to intercostal girder	7(8g)	25K	
Open floors—Brackets	7(5c)	. . .	
Over-lap on brackets	7(5c)	. . .	
"W"	7(5c)	. . .	
Strut	7(5c)	. . .	
"N"	7(5b)	. . .	
"l"	7(5b)	. . .	
Frame and Reverse	7(5b)	5	
Inner bottom—Center-line	7(6b)	4	
Margin	7(6e)	4	
Remainder	7(6a)	4	
Margin angle	7(6f)	4	
Intercostal girder	7(8)	4	
Top angles	7(8f)	3 notes	
Bottom angles	7(8e)	3 notes	
Hold frame, heel bracket (flanged)	7(7)	. . .	
Height of heel bracket	8(3)	. . .	
Gusset	7(7c)	. . .	
Approximate "l"	8(4a)	. . .	
Approximate "h" (to load line)	8(4a)	. . .	
$M = s \times h \times l^2 \times 0.01 \times 90$ percent	8(4a) and (4c)	. . .	

Item	Section	Table	Size
ε	8(4b)	. . .	
b	8(4b)	. . .	
K = s × b × c × 0.01 × 90 percent	8(4b) and (4c)	. . .	
Hold frame amidship	8(4)	6	
Over-lap on bilge bracket	8(3)	. . .	
Connecting angle to margin plate	7(7b)	. . .	
Second deck beams, etc.		4	
"l" outboard; middle	10(2a)	. . .	
"N" = s × c × h	10(2b)	. . .	
Through beams	10(2)	5	
Half beams	10(2)	5	
Girder (assume 20′ stanchion spacing)			
"M"	11(3b)	. . .	
Girder	11(3b)	10	
(*Note:* In practice any girder of the same total S.M. could be substituted)			
Stanchion "h"	11(2b)	. . .	
"W"	11(2b)	. . .	
"l"	11(2a)	. . .	
Stanchion size	11(2)	12	
Upper deck beams, etc.			
"N"	10(2b)	. . .	
Beam and girder same as second deck			
Stanchion "h"	11(2b) and 10(2b)	. . .	

Item	Section	Table	Size
"W"	11(2b)	...	
"l"	11(2a)	...	
Stanchion size	11(2)	12	
Transverse bulkhead. Assume 30″ spacing of stiffeners			
Hold stiffeners "N"	12(4c)	...	
(Bracketed) "l"	12(4c)	...	
Size	12(4c)	5	
'Tween deck stiffener "N"	12(4c)	...	
(Clipped) "l"	12(4c)	...	
Size	12(4c)	5	
Floor stiffener "N"	12(4c)	...	
"l"	12(4c)	...	
Size	12(4c)	5	
Plating. Write directly on Fig. 29	12(4a)	11	
Limber plate	12(4a)	11	
Floor plate	7(4c)	4	
Shell plating			
Freeboard to upper deck	...	...	
Table frame spacing	...	13	
Increase for actual spacing	15(2a)	13	
Table draft	...	14	
Increase for actual draft	15(2a)	14	
Shell plating—bottom amidships	15(2a)	14	
Shell plating—bottom forward	15(2a)	13	
Side shell, amidships	15(2a)	14	
Bottom and side at ends	15(3g)	13	
Sheer strake	15(2c, 3f)	14	

Item	Section	Table	Size
Freeboard to bridge deck	...	...	
Bridge side plating	17(1a)	14	
Upper deck			
Gunwale angle	15(5)	...	
Area clear of bridge	16(2)	14	
Area in way of bridge	16(2)	13	
Plating—minimum	16(4a)	15	
Plating a reast 20′ hatch	...	...	
Stringer plate—minimum	...	13	
Second Deck			
Area clear of bridge	16(2)	13	
Area in way of bridge	16(2)	13	
Thickness, minimum	16(4a)	15	
Thickness in way of 20′ hatch	...	...	
Stringer	16(3)	13	
Bridge Deck			
Area	16(2)	14	
Stringer, minimum	16(3)	13	
Plating, minimum	16(4a)	15	

There are many subjects in these rules which we have not covered: Hatchways, Section 18; riveting, Section 25; equipment, Section 24, etc., but we have gone far enough to see how the rules are used.

These rules are not invariable. If in a given case the authorities can be shown that a departure from the rules is justified, they will permit modification.

A wide variety of midship sections of various kinds of ships will be found in published descriptions of ships in marine magazines.

Chapter VI

WEIGHT CALCULATIONS

References

(*a*) "Principles of Naval Architecture," Volume I, Chapter III, Section 2.

(*b*) Lovett's "Applied Naval Architecture," Chapter 3.

(*c*) Kari, "Design of Merchant Ships," Part I, pages 29 to 40; all of Part II.

(*d*) Mackrow's "Naval Architect's, Shipbuilder's and Marine Engineer's Pocketbook."

The design of a ship depends on a correct estimate of the weight of the ship. In the first stages, the weight can be only roughly approximated; as the design takes definite form, the weight estimate is revised and made increasingly accurate.

The designer is dependent on accumulated data. At first the entire weight must be proportioned from similar ships already built. Later, some parts of the weight, such as the main steel items, can be fairly accurately calculated, but other parts, such as joiner work, remain incalculable until the working drawings are made, and must throughout the design be proportioned from weights of similar parts of previous ships. For this reason a naval architect, or the design department of a shipyard, must collect, classify and preserve finished-weight data.

For design purposes, a ship's weight is subdivided about as follows:

Steel: Main Hull. Erections. Houses and masts.

Miscellaneous Hull Weights:
Carpenter and joiner work. Floor coverings.
Cementing, paint, and bitumastic coverings.
Rigging and canvas. Insulation.
Furniture and steward's outfit. Boats and rafts.
Deck and navigating outfit. Anchors, chains and lines.

Hull Engineering: Deck machinery. Steering gear.
Ventilation. Pumping and drainage systems.
Refrigerating plant. Electric plant.
Wireless outfit. Interior communication.

Propelling Machinery: Engine room weights.
Boiler room weights. Shaft alley weights.

Designer's Margin (**to suit nature of estimate**)

Sum of above items is the. . . . . Light ship

Deadweight:
Fuel. Water. Passengers and crew.
Baggage, mail, and stores. Cargo.

Total deadweight

Displacement:
Light ship plus deadweight is the

Total displacement

The degree of detail (and the accuracy) possible or desirable in a weight estimate will depend on the stage of the design. It will vary from, at one extreme, the first "stab" as a coefficient times the deadweight, which may prove to be 15 percent wrong, to, at the other extreme, a complete detailed calculation from working drawings, with an error of perhaps 2 percent. The usual design weight estimate will be made in about four stages, somewhat as follows:

1. Trial displacement from deadweight-displacement ratio.
2. Estimate using "cubic numbers" from data.
3. Estimate using weights per square foot.
4. Final design weight.

First: The entire weight of the ship, or "light ship," will be estimated as a coefficient times the required deadweight. This coefficient will vary from about 0.35 for a

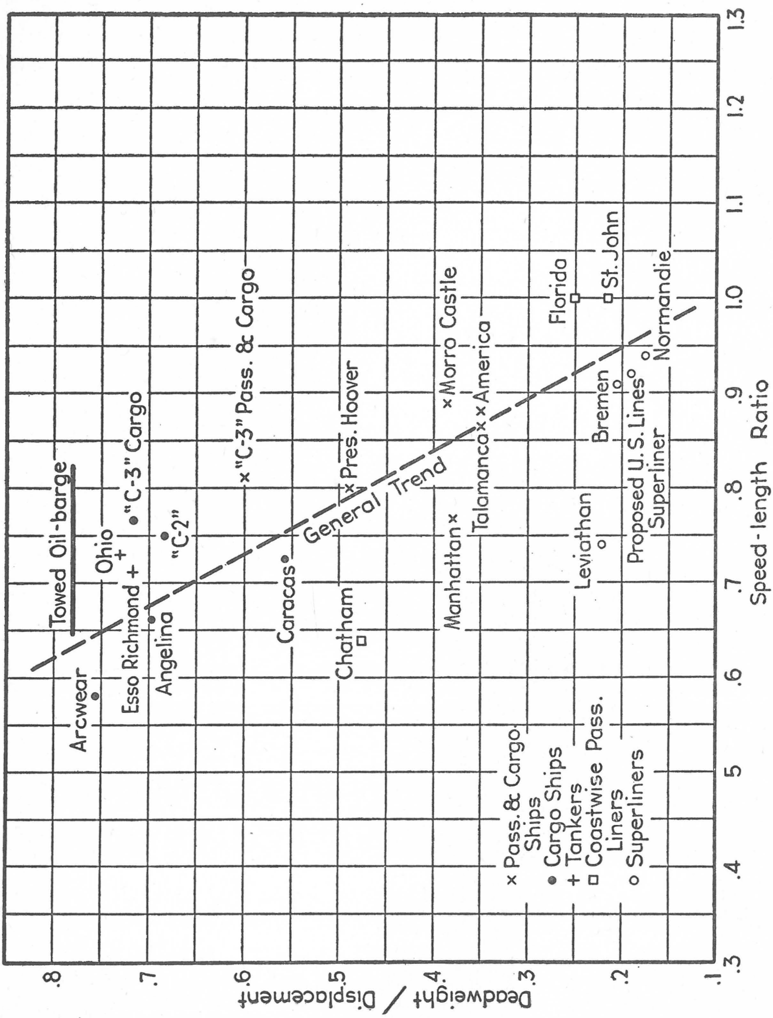


Fig. 30

slow, bare freighter, to about 4.0 for a fast de-luxe Atlantic liner. For data see Fig. 30, also Reference (*b*), pages 119 and 120, and Reference (*e*), pages 36 and 37. These data are given as a ratio of deadweight to displacement instead of

a ratio of light ship to deadweight, but since light ship plus deadweight equals displacement, if one ratio is known, the other can be found.

This coefficient varies widely with the character of the ship. It is reduced by increasing speed-length ratio, depth-draft ratio, relative number of passengers, elaborate specifications, etc. For a proposed ship it must be chosen from a closely similar ship.

Then will follow preliminary dimensions and powering, as will be shown in Chapter XIII.

Second: A somewhat better weight estimate will then be made as follows:

(*a*) Steel as a "cubic number coefficient" (see below) times $L \times B \times D/100$.

(*b*) Such miscellaneous hull items as vary with the size of ship (as "cementing," etc.) proportional to $L \times B \times D$.

(*c*) Such miscellaneous hull items as vary with the number of persons on board (as "furniture and steward's outfit") proportional to number of persons on board.

(*d*) Hull engineering proportional to $L \times B$.

(*e*) Propelling machinery proportional to *shp* (see below).

(*f*) Fuel oil and feed water proportional to *shp* and required cruising radius (see below).

(*g*) Passengers, crew and baggage proportional to persons on board.

(*h*) Stores and fresh water proportional to persons on board and cruising radius.

(*i*) Cargo.

The total of the above groups is the revised displacement.

In (*a*), the "cubic number coefficient" is the ratio

$$\frac{\text{weight in tons}}{L \times B \times D \div 100}$$

where L is length between perpendiculars, B is molded beam, and D is depth to the upper continuous deck. The "100" is to get a more convenient ratio. When applied to the

weight of the main steel hull (up to and including the upper continuous deck), it is a fairly constant and useful coefficient, varying from 0.25 to 0.32, except for tankers where it may be from 0.30 to 0.34. When applied to total steel it is still a useful coefficient, varying from 0.3 to 0.4. It can even be applied to total hull (light ship excluding machinery) varying from 0.4 to 0.6 and to be used with caution. Its greatest use, however, is when applied to steel weight only.

Corrections to Cubic Number Coefficient. As applied to main steel hull, this coefficient should be corrected for any difference in block coefficient between the type ship and the new ship by the factor:

$$\frac{1 + \frac{1}{2}\text{ new block coefficient}}{1 + \frac{1}{2}\text{ old block coefficient}}$$

It should also be corrected for any difference in length-depth ratio by the factor

$$\frac{\text{old } L/D}{\text{new } L/D}$$

Both factors are from Reference (*c*), pages 145 and 152.

Items (*b*), (*c*) and (*d*) vary too much to permit giving any data that would be of use.

(*e*) *Propelling Machinery.* At this stage of the design, the naval architect must make his own estimate of machinery weights. In the "final design weight," described later, the engineering department would furnish the weight of engineering items.

Propelling machinery weight varies with power, but is not proportional to power, as large installations will be lighter per horsepower than small installations. The weight will also vary with the type of machinery. See Fig. 31, also Reference (*c*), pages 34 to 39, and Reference (*d*), page 390, for data. It will be noted from Fig. 31 that reciprocating machinery with Scotch boilers will give from 4 to 6 indicated horsepower per ton for installations ranging from 2000 to 5000 indicated horsepower and that either geared turbines or turbo-electric installations with watertube boilers

will give from 7 to 13 shaft horsepower per ton for installations ranging from 4000 to 30,000 shaft horsepower. Very

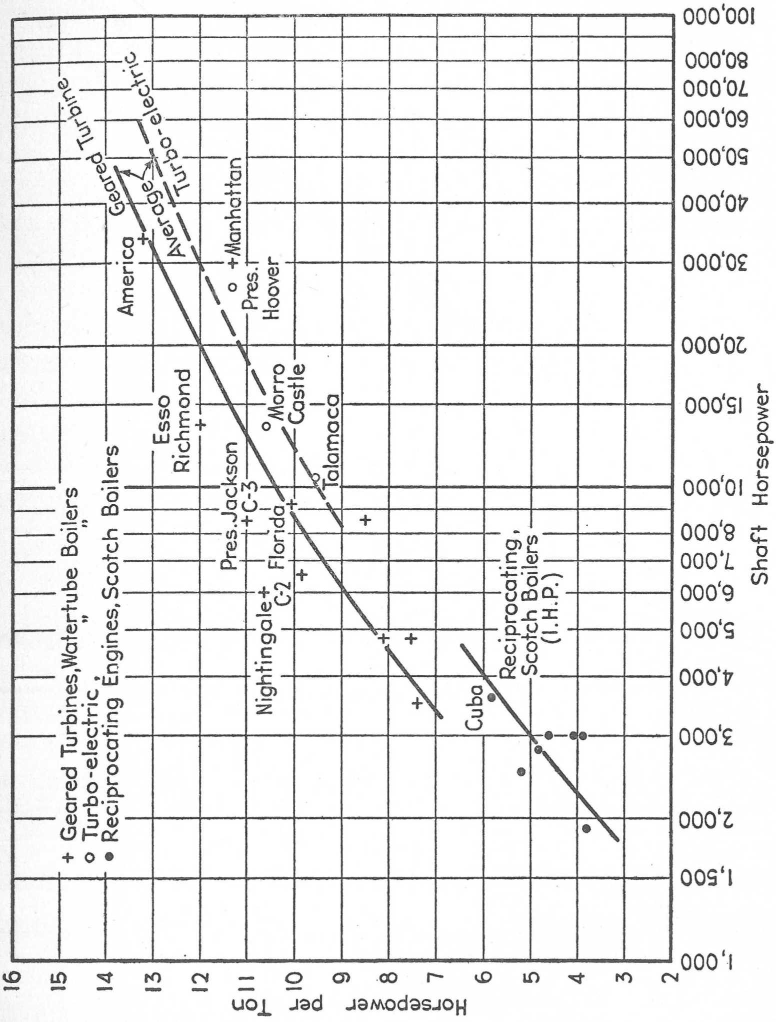


Fig. 31

large installations will be much lighter. For instance, the machinery of the U. S. Lines superliners proposed in 1930 was of about 200,000 shaft horsepower, and produced 22.7 shaft horsepower per ton for geared turbines and 21.8 shaft

horsepower per ton for turbo-electric machinery, while naval design for large ships may give 45 shaft horsepower per ton of propelling machinery. Data from strictly comparable actual ships are needed for preliminary design.

(*f*) *Fuel and Feed Water.* The required weight of fuel is found by dividing the required cruising radius by the speed in knots to get the hours of running, multiplying this by the shaft horsepower to get the shaft-horsepower-hours and using about 0.6 pound of oil per shaft horsepower per hour for steam machinery or about 0.4 pound per brake horsepower per hour for Diesel machinery. These rates will vary with the particular type and efficiency of the proposed machinery. To the amount of fuel so obtained there should be added a reserve of 15 percent for delays, heavy weather, etc.

The feed water could be proportional to the horsepower-hours, but in practice varies widely. If taken as 15 percent of the fuel, the amount will be reasonable.

(*g*) and (*h*) are small items; (*g*) will be about 0.1 ton per person, and (*h*) will be about 0.2 ton per person per day. (*i*), the cargo, is the only remaining item, and is assumed to have been given in the owner's requirements.

After this first revision of the weight estimate, the preliminary displacement, dimensions and power will be revised to suit the revised weight. The freeboard and stability will be checked, and if necessary the depth and beam modified to make these satisfactory. This will be shown in Chapter XIV. A sketch profile will be made, showing the number of decks, bulkheads, approximate extent of erections and houses, and the extent of cargo and fuel spaces, all approximately to the required volumes of cargo, fuel, machinery spaces, etc.

Third: A more accurate weight estimate can now be made, using approximate areas of shell and framing, decks, inner bottom, and bulkheads, with weights per square foot from ships already built. The other steel items, such as stem and stern, foundations, etc., must be proportioned from previous ships. The erections (poop, bridge and forecastle) will be proportional to (length of erections $\times$ B)

from previous ships, and the houses proportional to their volume. A similar degree of detail is for the first time possible in some of the miscellaneous hull weights. For instance, the hold ceiling and sparring, and carpenter decks, can now be approximated on a weight per square foot basis. The machinery and fuel weights will be revised to suit the revised power.

Areas. At this stage the area for shell and framing may be taken as $(B + 2D) \times L \times$ a coefficient from previous ships varying from 0.85 to 0.90. The decks can be taken as (number of decks $\times L \times B) \times$ (waterplane coefficient $+0.03$). The inner bottom can be taken as $L \times B \times$ (block coefficient -0.20). This is very approximate. The area of bulkheads can be taken as number of bulkheads $\times$ $B \times D \times$ (block coefficient $+ 0.05$).

Weights per Square Foot. These increase with the size of the ship. The following examples are typical (the weights are in pounds per square foot):

Length, load waterline	365′	382′	404′
Shell and framing	41.8	44.7	50.6
Inner bottom	22.6	25.2	27.0
Decks	21.1	25.7	28.4
Bulkheads	24.8	24.9	25.3

These are calculated detail weights for the entire structure, divided by the *gross* area (no openings deducted). Additional square foot weights for various structures are given in Reference (*c*), page 147.

After this weight revision, the displacement, dimensions and power are revised for the last time, and *lines* are drawn, as will be taken up in Chapter XV. A midship section showing the principal scantlings is prepared, and perhaps profile and deck plans showing scantlings. Arrangement plans and specifications are prepared.

Fourth: The final "design weight" can now be made, and the only limit to the amount of detail is the time available and the importance of the job.

A "weight per foot amidships" [see Fig. 32, and Reference (*c*), page 147] is calculated from the midship section. This is the weight of a section of the ship one foot long, amidships, and includes shell plating, keel, longitudinal framing, transverse framing, inner bottom, and deck beams and plating. The weight of a transverse member, such as a frame or deck beam, must be divided by the spacing in feet to get the weight per foot. Members which are lightened by holes, are reduced by the ratio of a lightened panel to an unlightened panel; this ratio will average about 85 percent. If there are longitudinal bulkheads which run for more than half the length between perpendiculars, as in an oil tanker, they are included in the weight per foot amidships.

Then the weight per foot amidships of each item is multiplied by the length between perpendiculars and by a longitudinal weight coefficient from previous similar ships, to obtain the total weight of the item. These coefficients are generally less than unity, due to the taper both of the ship and of the scantlings toward the ends. In the case of framing, the coefficient is often greater than unity, notwithstanding the taper. If time is limited, a single coefficient from the previous ship (usually between 0.75 and 0.85) may be applied to the entire weight per foot amidships to get the total weight of the corresponding items.

These "weight per foot amidships" coefficients are exceedingly useful. If we have from a previous similar ship (see "finished weights," later) accurate "weight per foot" coefficients, and from the midship section of the proposed ship accurate "weights per foot" of the proposed ship, we can estimate the weight of the items involved to probably within 3 percent, and these items constitute over half of the steel.

A typical calculation sheet is shown in Fig. 32. The weight per foot amidship (B) is multiplied by the longitudinal weight coefficient $A/(B \times L)$ (from previous ships) and by the length of the ship to get the total weight A.

Date________ **WEIGHT OF STRUCTURAL STEEL** P. S.______ No.______

For "Example Ship." Type Three-Island Freighter

Name________ Classification A.B. + A1

Nos.	ITEMS	Total Weight in Lb. (A)	Wgt. Per Ft. Mid. Sect. (B)	$\frac{A}{B \times L}$
1	Trans. Framing in Double Bottom	332000	1130	0.70
2	Trans. Framing Outside Double Bottom	403000	800	1.20
3	Long. Framing in Double Bottom	106000	230	1.10
4	Long. Framing Outside Double Bottom	—		
5	Shell Plating, Keel, Liners, etc.	1216000	3150	0.92
6	Double Bottom Plating, Margin, etc.	317000	1080	0.70
7	Oiltight Centerline Bulkhead	—		
8	Orlop Deck Plating, Beams, etc.	—		
9	Lower " " " "	—		
10	Main " " " "	—		
11	Upper " " " "	423000	1460	0.69
12	Second " " "	367000	1180	0.74
13				
14				
15	Expansion Trunk Bulkheads			
	TOTAL No. 1 to No. 15	3164000	9030	0.833
16	Stem, Sternframe, Struts, etc.	39000		
17	Main Traverse Bulkheads, W. T. and O. T.	261000		
18	Misc. Bulkheads, Inc. Chain Locker, etc.	81000		
19	Engine and Boiler Casing	—		
20	Engine and Boiler Foundations	62000		
21	Auxiliary Foundations (Inc. Deck Mach.)	17000		
22	Shaft Alley	62000		
23	Stanchions and Girders	99000		
24	F. W. Tanks and Foundations			
25	Vent. Trunks and Ducts	22000		
26	Cargo Hatch Coamings	232000		
27	Bilge Keel	15000		
28	Misc. Steel	98000		
29	Rivets and steel Tolerance	280000		
30				
31				
32				
33			Tons	
	TOTAL INTERNALS	1273000		Coeff. On (B)
	TOTAL STEEL TO DECK (C)	4437000	1980	1.17
	TOTAL ABOVE DECK	829000	370	
	TOTAL STEEL (D)	5266000	2350	1.39

DIMENSIONS: Length (L) 420' Beam 54' Depth 30'-3"

Cubic No. 6860 Steel Coeff. On (C) 0.289 Steel Coeff. On (D) 0.343

Remarks:

Fig. 32

The items other than midship section items are then estimated. The stem, etc., are taken from similar ships. One complete watertight bulkhead may be worked out by American Bureau of Shipping rules as was done in Chapter V, its total weight calculated and divided by its area to get its weight per square foot. This weight per square foot must be increased by about 10 percent (to allow for increased strength requirements for deep-tank bulkheads, peak bulkheads, etc., and for local strengthening of various kinds) to get the average weight per square foot of all the bulkheads in the ship. Or, if available, actual square-foot weights of bulkheads on a similar previous ship may be used. The average weight per square foot so obtained is multiplied by the total area of all the bulkheads in the proposed ship to get the weight of bulkheads. The casings, foundations and other miscellaneous steel items are proportioned from similar ships, modified as seems proper. The erections (poop, bridge, forecastle) are calculated from the steel scantling plans. The house steel is obtained as areas times weight per square foot.

For large important ships weights obtained by "weight per foot" amidships and coefficients may not be good enough. In such cases curves of weight per foot may be used. For instance, if the weight per foot of the shell plating be obtained, not only at amidships but at sufficient points along the vessel's length to permit plotting a curve of weight per foot, the area under this curve, obtained either by using Simpson's rule or by planimeter, will give the weight of the shell plating. Decks, inner bottom, and longitudinal bulkheads can be done the same way. Butt laps must not be forgotten. Framing can be done similarly, but will not be one smooth curve, because of changes in type of construction.

The miscellaneous hull items are calculated from the arrangement plans, specifications, and returned weights from previous ships. Areas of carpenter work, joiner work, insulation, deck covering, etc., are obtained from the arrangement plans and given suitable weights per square foot.

Furniture and outfit are listed from the specifications and plans and given suitable unit weights, and so on. The hull engineering and machinery weights are similarly worked up by the engineering department.

It is evident that the calculator is dependent, from start to finish, on weight data from previous ships, from the broad coefficients used in preliminary work, to the detailed unit weights of the various articles listed in the final detail calculations of "design weight." These data are obtained by either calculating or weighing "finished weights" of actual ships.

Finished Weights

Finished weights of previous ships, which have been shown to be indispensable to the designer, come from two sources. Either the weight is calculated in detail from the working drawings, or all the material entering into the ship is weighed. For naval vessels the latter is usually done.

There are differences of opinion as to which is preferable. A weighed weight should be more accurate than a calculated weight, but this is not necessarily so. Some items where much of the work is done on board, as joiner work, insulation, etc., cannot be weighed finished; the material going aboard and the scrap coming off must be weighed, and the classification of this scrap into the desired weight groups is nearly impossible. Above all, the value of returned weights depends on the attitude of the men obtaining them. If, as is natural, they feel that the weighing of materials is a somewhat useless proceeding that delays the real objective of getting the material into the ship, the result is liable to be undependable. Once a weight gets past the weigh-master and worked into the ship, it cannot be weighed. The first requisite of returned weights is dependability; without it they are worthless, for in general they cannot be checked.

The calculator can make mistakes too. He must be sufficiently familiar with ship's structure to read between the lines of a drawing and include items not detailed, such as

hangers for piping and false work behind joiner ceilings. But he is not pressed by the necessity of getting the work into the ship. Also, he can include with his weight an estimate of the center of gravity, which the weigh-master cannot do. A calculated finished weight, conscientiously done, is a long, tedious job, but the result is in general of more value to the designer than a returned weight.

Center of Gravity

When a finished weight is calculated, a *VCG* and *LCG* are calculated at the same time. Heights are taken above the base line, and longitudinal distances from either amidships or the forward perpendicular. Using the forward perpendicular eliminates the possibility of error involved in using forward and after distances. The final result can be changed to a distance from amidships if desired.

A returned weight gives no information as to the center of gravity, and must be broken down as necessary to permit writing centers for individual items and calculating the combined center.

For preliminary work the center of gravity must be proportioned from calculated centers of previous ships. Vertical centers can often be proportioned to the depth of the ship, and longitudinal centers to the length of the ship. In other cases centers can be estimated with fair accuracy from the sketch profile made early in the design.

When using "weight per foot amidships", a close estimate of *VCG* of the "weight per foot" items can be obtained by having coefficients of $\frac{VCG \text{ of total item}}{VCG \text{ of same item amidships}}$ from the data ship. Then the *VCG* of each item of the "weight per foot amidships" can be calculated for the new ship from the midship section and multiplied by these coefficients to get the *VCG* of the entire weight of each item.

It is usual to allow a margin of 0.5 foot or so in the estimated *VCG* of a proposed ship to allow for the tendency of the final *VCG* to be higher than was estimated.

Problems

1. An owner wants a ship of the same general character as the example ship (see page 12) but to carry 10,000 tons deadweight. (a) What deadweight coefficient, and (b) what approximate displacement would be used?

2. Using the weights given on page 13, what is the "cubic number coefficient" of the example ship, based (a) on main hull only, and (b) on total steel? (c) What would be approximately the main hull weight of a similar ship 450 feet between perpendiculars × 62 feet beam × 34 feet deep?

3. A proposed ship is tentatively 500 feet between perpendiculars, 65 feet beam, 36 feet depth. The weight per square foot of shell and framing (the first five items in Fig. 32) for the most similar ship available is 52 pounds. What will be the approximate weight of shell and framing on the proposed ship? Use an area coefficient of 0.88.

4. Using the scantlings written in on the midship section (Fig. 28) of the example ship, calculate the weight per foot amidships of the transverse framing within the double bottom. Assume that 50 percent of the floors are solid and 50 percent open. Include clips on floors to center and side girder and to margin plate.

0.36-inch plating weighs 14.7 pounds per square foot.
3-inch × 3-inch × 7⁄16-inch angle weighs 8.3 pounds per foot.
3-inch × 3½-inch × 7⁄16-inch angle weighs 9.1 pounds per foot.
7-inch × 2.95-inch × 0.325-inch channel weighs 16.4 pounds per foot.
6-inch × 3½-inch × ⅜-inch angle weighs 11.7 pounds per foot.

5. (a) What would be the approximate weight of geared-turbine propelling machinery of 10,000 shaft horsepower (use Fig. 31)? (b) What weight should be allowed for fuel oil for this machinery for a cruising radius of 9000 miles, assuming the speed of the ship to be 18 knots?

Chapter VII

WAVES AND ROLLING

References

(*a*) "Principles of Naval Architecture," Volume II, Chapter I.
(*b*) Attwood's "Theoretical Naval Architecture," Chapter VI.

The study of waves can lead to involved mathematical analysis, which will be omitted here. The naval architect needs actually to know comparatively little about waves. He is interested in them for three reasons:

(a) They cause rolling, which, if excessive, can cause severe racking stresses, discourage passenger traffic and damage fragile cargo.

(b) They bend the ship girder, causing stresses which must be designed for, as will be discussed in Chapter VIII.

(c) If large, they reduce the speed of the ship.

Mathematicians have developed the trochoidal theory of waves. All observed data as to contour, speed, length and period of waves conform so closely to this theory that it is generally accepted. The fundamental assumption is that every particle of water travels in a circular orbit at constant velocity, and that the profile, not only of the wave surface but of every sub-layer which would have been horizontal in still water, is a trochoid (see "Definitions," Chapter I) in which the rolling circle rolls on the underside of the straight line. The trochoid formed by the surface is that traced when the circumference of the rolling circle is the length of the wave, and the radius to the tracing point is half the height of the wave from hollow to crest. The crest is a little sharper than the hollow.

The following formulas result from this theory:

Let

L = length of wave, in feet.
H = height of wave from hollow to crest, in feet.
V = speed of wave, in knots.
T = period of wave, in seconds.
v = orbital velocity of water at surface, in feet per second.
E = energy of one wave per foot of width, in foot-pounds.
s = sine of slope of steepest part of wave profile.
a = vertical acceleration at crest or hollow of wave.
g = acceleration due to gravity.

Then

$L = 5.12T^2 = 0.557V^2$.
$T = 0.442\sqrt{L} = 0.330V$.
$V = 3.03T = 1.34\sqrt{L}$.
$v = 7.11H/\sqrt{L}$.
$E = 8LH^2$ (in salt water of 64 pounds per cubic foot).
$s = \pi H/L$.
$a = g\pi H/L$.

Length, period and speed depend upon each other and are independent of height.

Size and Proportion of Waves

The length and height of waves resulting from a long-continued wind depend on the wind velocity and the expanse of water. North Atlantic waves seldom exceed 600 feet in length. The maximum observed height of short waves is relatively larger than that of long waves, varying from about $L/15$ for 300-foot waves (20 feet high) to $L/25$ for 1000-foot waves (40 feet high). For the standard strength calculation discussed in Chapter VIII, a standard wave whose length is that of the ship and whose height is one-twentieth of the length is used. This is about right for

ships from 400 to 500 feet long, but is not high enough for smaller ships and is too high for larger ships. This is taken care of by varying the permissible stress, as discussed in Chapter VIII.

The ordinates of a trochoid vary with the ratio of length to height, and are tabulated here for three ratios. The distance from hollow to crest is divided into 10 spaces.

	Hollow	1	2	3	4	5	6	7	8	9	Crest
$L/H = 15$	0	0.017	0.066	0.148	0.260	0.398	0.552	0.711	0.856	0.962	1.000
$L/H = 20$	0	0.018	0.073	0.161	0.280	0.423	0.579	0.733	0.872	0.966	1.000
$L/H = 25$	0	0.019	0.077	0.169	0.290	0.437	0.596	0.748	0.880	0.970	1.000

Effects of Rolling

If the ship's rolling period synchronizes with the period of the waves, the ship will roll badly.

It was shown, page 45, that a normal ship will roll with a period of about $1.9\sqrt{B}$, or from 14 to 18 seconds. For this period to equal the period of broadside waves, the length of the waves must be (see page 91) from 1000 to 2500 feet long. Normal ships are evidently not likely to get into synchronism with broadside waves.

When waves are not broadside, but are either meeting or following the ship at an angle, their *apparent* period is not $0.442\sqrt{L}$. If meeting, the period is quicker; if following, slower. In the latter case it may be impossible to avoid synchronism except by changing the course or the speed.

The formula for the pitching period is the same as for the rolling period with longitudinal values of K and GM substituted for transverse values (see page 44). Longitudinal K will be about 0.25 LBP. The pitching period is roughly half the rolling period. In a short head sea, the pitching period of the ship is slower than the apparent period of the waves, and the ship will drive through them with but little pitching. In a long head sea, or a following sea, the pitching period of the ship may be quicker than the ap-

parent period of the waves, and the ship will ride each wave with perhaps considerable vertical movement, but fairly dry decks. In between is synchronous pitching, which may be violent even in a flat swell and will require a change in course or speed to get out of synchronism.

Acceleration at End of Roll

At the end of a roll, when the ship is starting to return to the upright, there is an acceleration of every weight in the ship, acting at right angles to a line from the weight to the axis of rolling (see Fig. 33). The axis of rolling is near the

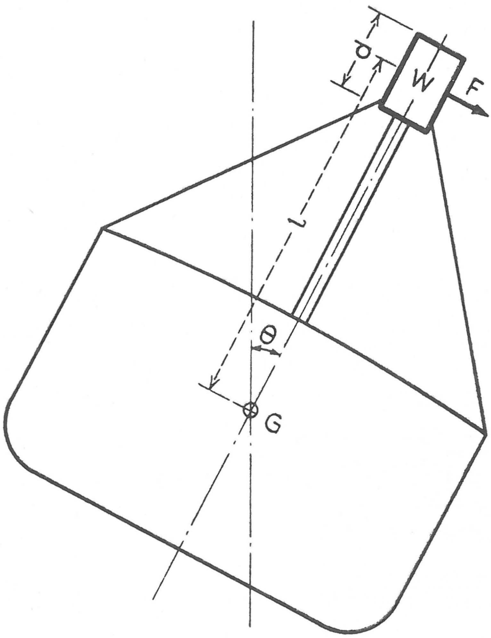


Fig. 33

center of gravity, and is usually taken to be at that point. The tangential force required to produce this acceleration becomes important for weights a long distance from the axis of rolling. This force has caused sailing ships, which

were necessarily "stiff" and rolled quickly when not under sail, to "roll their masts out."

This force can be shown to be [see Reference (*a*) or (*b*) and Fig. 33]

$$F = \frac{W}{g} \times \frac{4\pi^2\theta l}{T^2} \times \frac{\pi}{180} = 0.0214\,\frac{W\theta l}{T^2}$$

where

F = tangential force, in pounds.
W = weight, in pounds.
θ = maximum angle of roll, in degrees.
l = distance from axis of rolling.
T = period of complete (double) roll.

Strictly, l is not to the center of gravity of W, but to the radius of gyration of W about G. If d (in Fig. 33) is large in relation to l, this cannot be neglected. In such cases use, instead of l, $l' = \sqrt{l^2 + d^2/12}$. If d is less than 0.49 l, the error involved in neglecting this correction is less than 1 percent.

In cases involving this lateral force, two other forces acting in the same direction are usually present; the lateral component of the weight due to the angle of inclination, and a horizontal pressure from the wind naturally associated with heavy rolling.

The lateral component of the weight is $W \sin \theta$.

The wind pressure will be from 10 to 30 pounds per square foot of area exposed to the wind, depending on assumed conditions. It varies with the shape of the area. Ten pounds is approximately the pressure in a 60-mile wind and is an average figure for merchant ship calculations. On naval vessels, a 30-pound pressure, in conjunction with a 30-degree roll is often used; 30 pounds corresponds to a wind of about 110 miles per hour.

The preceding formula is strictly true only for isochronous, unresisted rolling. Isochronous means having the same period regardless of amplitude of roll. Unresisted means

not retarded by the resistance of the water. Neither condition is strictly true of actual rolling; but, if the corrections be omitted, the result is sufficiently accurate for practical purposes.

Apparent Weight on a Wave

The apparent weight of a body which is given a vertical acceleration is greater or less than its actual weight, according to whether the acceleration is upward or downward. This is a familiar sensation to a child on a swing or on a roller coaster. The water in a wave, or an object floating on this water, is heavy when at the bottom of a wave and light when at the crest.

Two results of this are of interest; one to the naval architect, the other to the small-boat sailor.

(1) The standard longitudinal strength calculation (see Chapter VIII) ignores this effect. The actual increase in buoyancy at a crest and loss of buoyancy in a hollow are both less than are assumed, and the actual bending moment is less than that calculated. This is the "Smith correction," discussed on page 240 of Reference (*b*), and has been calculated to amount to about 5 percent. In practice this correction is *not* made.

(2) The other effect is the loss of "power to carry sail" when on a crest. When a vessel is inclined, the righting moment is $\Delta \times GZ$. On a crest, the apparent Δ is reduced. If the inclination were due to a weight on the boat, that weight also would apparently decrease, and angle of heel would not change. But if the inclining moment is due to wind, it would not decrease, and the angle of heel would increase. If the vessel were already heeled well over, the increase might result in a "knock-down."

Numerically the apparent change in weight equals $(W/g) \times a$. Since a equals (page 91) $g\pi H/L$, the apparent change in weight (increase in a hollow, decrease on a crest) = $W\pi H/L$.

Effect of Waves on Speed

Broadside waves have but little effect on speed. A head sea always reduces speed. A following sea may increase speed slightly unless it is associated with heavy pitching.

The reduction in speed due to a head sea is greatest in waves of about the ship's length. The apparent period of such waves is not far from that which would synchronize with the ship's pitching period, and extremely violent pitching may result, bringing the ship nearly to a standstill at times. In much shorter, or much longer, waves, the reduction in speed is much less. There is not much that the naval architect can do about this, except that, if the vessel is expected to operate most of the time in waves much longer or much shorter than her own length, the forward lines will be modified accordingly. For short waves a fine waterline entrance and but little flare just above the waterline are best, since the ship must drive through the wave, while for long waves the reverse is better, to help the ship to rise with the oncoming wave.

Problems

1. (a) What is the speed of a wave 400 feet long and 20 feet high? (b) What is the orbital velocity of a particle of water on the surface?

2. The example ship is observed to be rolling with a period of 15 seconds. Approximately what is her *GM*? (See page 44 or Fig. 16.)

3. What length of broadside waves would synchronize with this vessel?

4. What would be the total lateral force on a weight of 500 pounds with an exposed area of 10 square feet, 100 feet from the axis of rolling on this vessel, at the end of a 30-degree roll, assuming 10 pounds per square foot wind pressure?

Chapter VIII

STANDARD LONGITUDINAL STRENGTH CALCULATION

REFERENCES

(*a*) "Principles of Naval Architecture," Volume I, Chapter VI.
(*b*) Attwood's "Theoretical Naval Architecture," Chapter VIII.
(*c*) Lovett's "Applied Naval Architecture," Chapter XII.
(*d*) Murray's "Strength of Ships."

This calculation consists of finding the bending stress in the hull girder when the ship is afloat across a "standard" wave, and is based on the following assumptions:

1. The length of the standard wave equals the ship's length, and the height of the wave is one-twentieth of the length.

2. The wave profile is a trochoid, dimensions for which are given in Chapter VII, on page 92.

3. It is usually assumed, and approximately true, that the maximum hogging stress occurs when a crest is amidships and a hollow at each end, and that the maximum sagging stress occurs when a hollow is amidships and a crest at each end.

In an elaborate investigation the bending moment may be obtained with the crest of the wave at a series of positions along the ship, and the real maximum found. In such cases the maximum hogging moment has been found to be from 0 to 1½ percent greater than when the crest is amidships, and the maximum sagging moment from ½ to 9 percent greater than when the hollow is amidships. The maximum bending moment nearer the ends of the vessel, however, will be considerably greater than with the crest or hollow amidships.

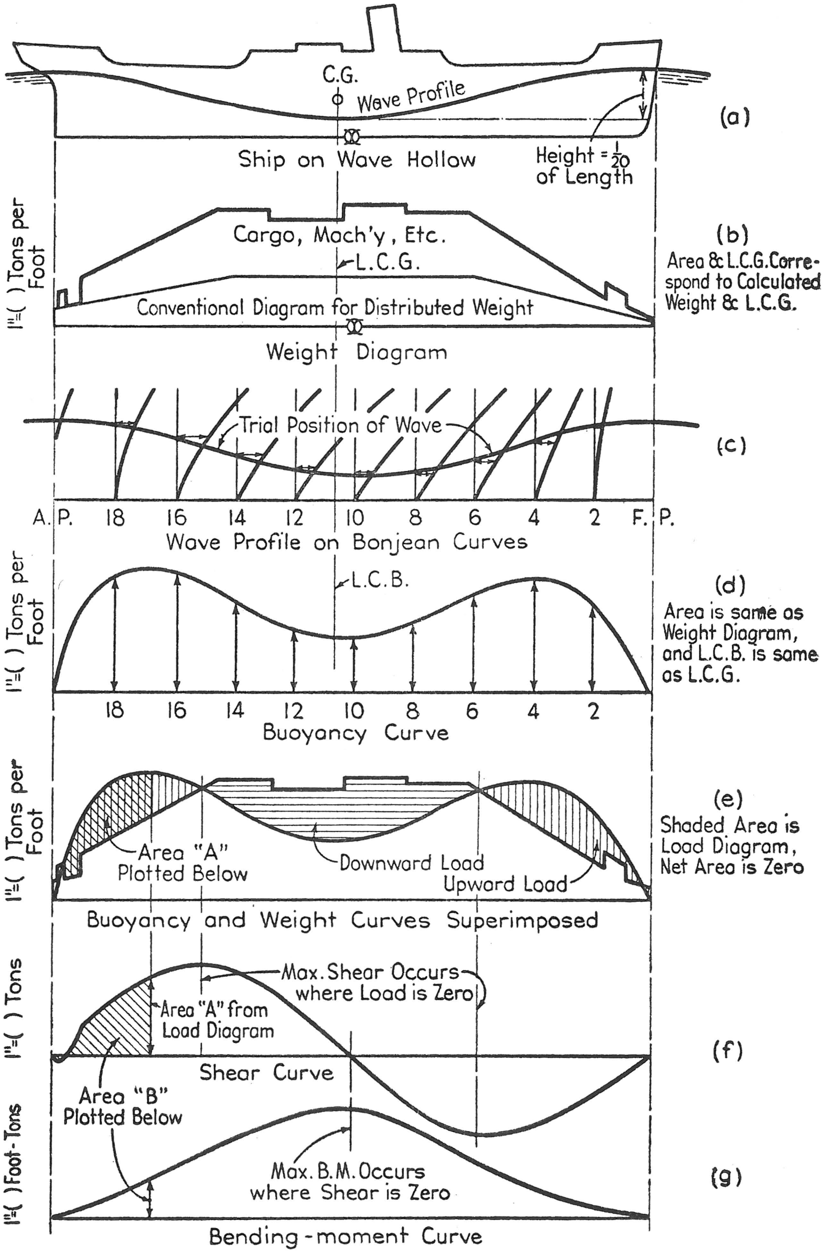


Fig. 34.—Illustrating steps in longitudinal strength calculation

The steps involved in the longitudinal strength calculation are as follows (see Fig. 34):

1. Weight and *LCG* of the Loaded Ship. It is usual to use the fully loaded ship for the sagging moment and to use the "end of voyage," or "burned-out" condition; that is, the fully loaded ship minus the fuel and water, which are usually near amidships, for the hogging moment.

2. Weight Diagram. See Fig. 34(b), and Reference (*a*), page 208. A weight diagram is drawn, whose base line represents the length of the ship, and whose height at any point represents the total weight per foot of the ship and its load at that point. The total area of this diagram must correspond to the weight of the loaded ship, and the *LCG* of the

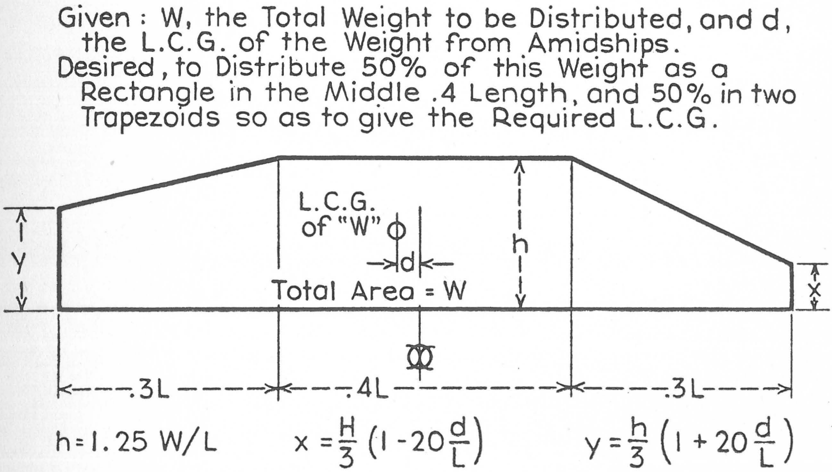
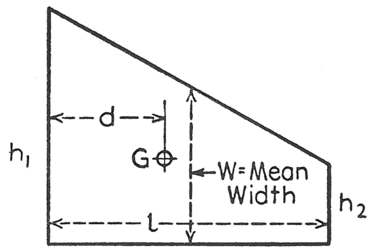


Fig. 35

diagram must correspond to the *LCG* of the loaded ship. Short cuts are necessary in drawing this diagram. A large amount of detail may be, and sometimes is, included, but does not change the final bending moment much. Most of the hull weights are usually drawn as a single area, as, for instance, a rectangle and two trapezoids, as on page 208 of Reference (*a*) and in Fig. 34(b). This grouped weight may consist of the steel, woodwork, fittings and outfit except

anchors and cables, hull engineering except windlass and steering gear, any spread-out items of deadweight, such as passengers and crew, and designer's margin. The diagram must be so proportioned that not only will the area be correct but also the *LCG*. This may be done using the ordinates given in Fig. 35.

The machinery weights are distributed over the length of machinery spaces, usually as rectangles. Fuel and water are distributed over the length of the tanks, as rectangles or trapezoids. A trapezoid may be drawn with any desired location of its center of gravity within its middle one-third length, by making the end ordinates as in Fig. 36.



$$h_1 = W(4 - 6\,d/l)$$
$$h_2 = W(6\,d/l - 2)$$

Fig. 36

Cargo, often the largest area of all, is distributed over the length of cargo holds as trapezoids, and so on until the diagram includes all the weights in the loaded ship. Usually, due to the accumulation of slight errors, the final curve will need to be slightly adjusted to give the required area and *LCG*. The center of the various partial areas and of the final total area is checked by using an integrator (see Chapter I) or, if none is available, by cutting out and balancing the diagram.

3. Curve of Buoyancy. See Fig. 34(c) and (d). The next step is to get a curve of buoyancy whose area and *LCG* are the same as those of the weight curve. Draw on a separate sheet of paper a standard wave profile to the same horizontal and vertical scales as were used on the drawing of

Bonjean curves (see "Curves of Form"). Place this wave profile under the tracing of the Bonjean curves in a trial position that looks reasonable, as in Fig. 34(c), read the area of each station at the height where the wave profile crosses the station, and compute the displacement and *LCB* as in "Form Calculations," Chapter II. The displacement and *LCB* will probably not be the required values, and it will be evident whether the trial position of the wave profile was too high or too low, and whether the bow or the stern should be immersed more deeply to move the *LCB* toward the required position. Further trial positions are calculated until the displacement and *LCB* are correct. The displacement per foot in tons, at each station, is the area (both sides) of that station divided by 35. Using these values of displacement per foot, a curve of buoyancy, Fig. 34(d), is drawn on the same sheet, and to the same scale, as the weight diagram. This curve, too, will usually require slight adjustment to make its area and *LCB* just right, by arbitrarily adding or taking off a thin sliver as necessary. The integrator (or balancing) is again used to check the *LCB* of the buoyancy curve.

The strength diagram will now look somewhat like Fig. 34(e). Where the buoyancy is greater than the weight, the net result is an upward load, and vice versa. A curve of loads may be drawn, but this is not necessary.

The vertical shear at any point is equal to the area of the load diagram (the area between the weight curve and the buoyancy curve) up to that point. Using a planimeter or the planimeter wheel of the integrator, get the area of the load diagram up to a sufficient number of points to plot a curve of shear, to any convenient scale, as in Fig. 34(f). Note that the slope, or steepness, of the shear curve is everywhere proportional to the width of the load diagram. This fact makes it possible to draw a correct curve of shear with a minimum of points.

4. **The Bending Moment** at any point is equal to the area under the shear curve up to that point. So, using the planimeter again, find the area of the shear curve up to a

sufficient number of points to plot a curve of bending moments, Fig. 34(g), to any convenient scale. Note again that the slope of the *BM* curve is everywhere proportional to the ordinate of the shear curve. The maximum bending moment will occur where the shear is zero, and will be the area of either lobe of the shear curve.

A planimeter constant must be derived for each curve to suit the scale used in constructing that curve.

5. Section Modulus. The hull girder of a ship is only an exaggerated case of a built-up section (see page 64) and its section modulus is calculated as such, except that it is customary to use vertical dimensions in feet instead of inches. Areas are in square inches. A convenient form for tabulating the calculation is shown in Fig. 37, which is self-explanatory except perhaps the last column (7). This column gives the moment of inertia of individual items about their own *CG* and is used only for vertical plates, such as shell, vertical keel, longitudinal bulkheads, if any, etc. The inertia of horizontal plates and of shapes about their own axes are negligible. *A* is the area of the plate in square inches and *h* is the vertical extent in feet. If the plate is inclined, as a bilge strake, *H* is the vertical projection of the width.

Note that columns (4) and (5) will be either plus or minus, but that column (6) is always plus. The actual neutral axis is found as shown, and the moment of inertia about the assumed axis corrected to the actual axis as shown. The distances from the neutral axis to the keel and to the deck will be different, and the stresses in the keel and deck will differ accordingly.

Tension stresses (in the keel for sagging, and in the deck for hogging) in riveted ships should be increased over the value obtained as above, for the effect of rivet holes, by the ratio of original area of metal to remaining area of metal between the holes. This ratio will vary in the various parts of the structure, and an average of 1.15 to 1.20 is used.

Continuous fore-and-aft members only are to be included in the section modulus. For instance, the intercostal side

Form for Calculating Moment of Inertia of Ship's Section

S. S.____________________

INERTIA CALCULATION TO________DECK

Axis Assumed at_____'Above B. L.

① Items	② Scantlings in Inches	③ Area in Sq. In.	④ Lever in Feet	⑤ Moment about Assumed Axis	⑥ Lever in Feet	⑦ Moment of Inertia about Assumed Axis	⑧ I_0 Correction $= \frac{1}{12}Ah^2$
Below Axis			−	−	−	+	
Above Axis			+	+	+	+	
Totals		Sum_1		Sum_2		Sum_3	Sum_4

Actual N. A. $\left(\begin{matrix}\text{Above}\\ \text{Below}\end{matrix}\right)$ Assumed N. A. = Sum_1 / Sum_2 = "d"

Moment of Inertia about Assumed N. A. = $Sum_3 + Sum_4$

I = Moment of Inertia about Actual N. A. = $(Sum_3 + Sum_4) - (Sum_1 \times d^2)$

Section Modulus to Deck = I ÷ (Distance from Actual N. A. to Extreme Top Fiber)
to Keel = I ÷ (" " " " " Bottom Fiber)

Fig. 37

longitudinal girder within the double bottom of the example ship (see Fig. 28) would not be included.

Permissible stress varies with the size of the ship. This, while apparently illogical, results from the facts that allowance for corrosion does not vary with size of ship, and that the standard wave is too high for long ships, but too shallow for short ships. Instead of using varying proportions of waves more in accord with actual storm waves, it is more convenient to use the standard wave and to vary the resulting permissible stress. This stress will be (in tons) approximately the cube root of the length in feet; that is, about 7 tons per square inch for a 400-foot ship and 10 tons per square inch for a 1000-foot ship.

Superstructure and Deck Houses

If several tiers of long houses are built of light scantlings above the strength deck of a ship, there is danger that y to the top extreme fiber will increase more than I and that the section modulus will actually decrease. Fractures of the superstructures may result, as in the case of the old *America* described in the *Transactions* of The Society of Naval Architects and Marine Engineers, 1927. Care must be taken in such cases either to keep the I/y of the combined structure at least equal to that of the hull alone (see American Bureau of Shipping rules, Section 17, paragraph 4) or to cut the superstructures at closely spaced intervals with expansion joints. The former is gaining favor, and several large liners, both in this country and abroad, have been built without expansion joints.

Without Expansion Joints	With Expansion Joints
Manhattan and *Washington*	*President Hoover* and *President Coolidge*
Monarch and *Queen of Bermuda*	*Rex*
Empress of Britain	*Bremen*
Mariposa, etc.	*Queen Mary*
Conti di Savoia	
America (new)	

Bending Moment Factors

The maximum bending moment in a ship on the standard wave has been found to be roughly proportional to (displacement $\times$ length) and for preliminary work is often taken as $(\Delta \times L) \div$ a factor. This factor is in the neighborhood of 30, varying from about 25 to about 35. See Reference (*a*), page 212; Reference (*c*), page 515, and Reference (*d*), page 180, for further data on bending moment factors.

Problems

1. Assume that the fully loaded condition of the example ship is made up as follows:

	Tons	*LCG* from *FP*
Light ship	3580	220′
Passengers, crew and stores	10	250′
Fuel oil	1050	188′
Feed water	210	290′
Drinking and culinary water	100	350′
Cargo	6750	205′
Total displacement	11,700	211′

What weight and *LCG* should be used for getting the hogging bending moment?

2. Assume that 2688 tons of the above weight, with a center 214.2 feet from the *FP*, is to be distributed as a rectangle and two trapezoids as in Fig. 35. Compute h, x and y.

Prove that the resulting area and *LCG* are correct.

3. Arranging the work as in Fig. 37, calculate (a) the section modulus of the rectangular box section shown in Fig. 38. Then calculate (b) the section modulus of the same section but with the sides carried up past the top, 2 feet.

Compare the two results, and notice the relation this problem bears to the discussion of superstructure in the preceding test.

4. Assume that (a) the hogging bending moment of the example ship, in "end of voyage" condition, is 120,000 foot-

tons, and the sagging bending moment, in "full load condition," is 150,000 foot-tons; (b) the moment of inertia of the midship section (excluding the bridge) is 340,000 square inches × feet^2 with a NA 14.0 feet above BL, both figures

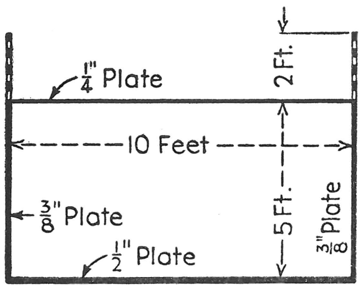


Fig. 38

ignoring rivet holes, and (c) rivets reduce the section by an average of 18 percent. Depth, keel to deck at centerline, 31 feet 3 inches.

Question: (a) What are the hogging and sagging bending moment factors?

(b) What is the amount and kind (tension or compression) of the maximum stress in the main deck and in the keel?

Chapter IX

POWER AND RESISTANCE, TAYLOR'S STANDARD SERIES

References

(*a*) Taylor's "Speed and Power of Ships."
(*b*) "Principles of Naval Architecture," Volume II, Chapter II.
(*c*) Attwood's "Theoretical Naval Architecture," Chapter IX.

Power

Power is the product of a force times the speed at which the force is exerted. To measure power, one must measure a force, a distance and a time. This can be expressed either as a force times a speed (as in effective horsepower and indicated horsepower below) or as a torque (that is, a twisting moment as in a shaft) and a rate of rotation, as in shaft horsepower.

Horsepower is the unit of power usually used in connection with ships. One horsepower is the power developed when the product of the force in pounds and the speed in feet per minute is 33,000; that is,

$$1\ hp = 33{,}000 \text{ foot-pounds per minute}$$

or

$$hp = (\text{force in pounds}) \times (\text{speed in feet per minute}) \div 33{,}000$$

Three kinds of horsepower are commonly referred to in ship work: effective horsepower, shaft or brake horsepower, and indicated horsepower, all defined in Chapter I.

Effective horsepower (*ehp*) is calculated from model tests and is not measured at sea, except rarely as a full-size laboratory experiment such as Froude's *Greyhound* experiments described on page 318 of Reference (*c*).

$$ehp = \frac{\text{(resistance in pounds)} \times \text{(speed in feet per minute)}}{33{,}000}$$

or

$$\frac{\text{(resistance in pounds)} \times \text{(speed in knots)}}{325.7}$$

Shaft horsepower (*shp*) is calculated from the torque in the shaft and the revolutions per minute. If the torque in foot-pounds is $\mathcal{Q}$, this torque can be considered as a force of $\mathcal{Q}$ pounds acting 1 foot from the center of the shaft. If the shaft is turning N times per minute, the distance traveled by this force per minute would be $2\pi N$ feet, and the power would be

$$shp = \mathcal{Q} \times 2\pi N \div 33{,}000$$

or

$$\mathcal{Q} \times N \div 5252$$

Torque is measured on the ship by a torsionmeter, which really measures the twist in a section of the shaft, or in the shop by a dynamometer brake (hence the term brake horsepower).

Indicated horsepower (*ihp*) is determined by measuring the pressure within the cylinders of the engine by an indicator. This device draws a curve whose base line represents the travel of the piston and whose ordinate shows the pressure at each moment of that travel. From this the mean effective pressure (P) is found. The average force exerted is PA, where A is piston area. The distance per minute is LN, when L is the length of stroke and N the revolutions per minute, so

$$ihp \text{ (for one single-acting cylinder)} = PLAN/33{,}000$$

Effective horsepower will be from 60 to 70 percent of shaft horsepower. Most of this loss is in the propeller, whose efficiency is seldom over 70 percent.

Shaft horsepower will be from 85 to 90 percent of the indicated horsepower of a steam reciprocating and from 70 to 80 percent for a Diesel engine. The loss is in the mechanical resistance of the engine.

Effective horsepower is the starting point, the others are simply the powers required at various points to obtain the required effective horsepower. This brings us to resistance.

Resistance

The resistance of a ship consists of three main parts: Frictional, wave-making and eddy-making (see "Definitions," Chapter I.)

Frictional resistance would be the only resistance for an extremely thin plank with no waves and no eddies. It is surprisingly large; for moderate-speed ships it is roughly two-thirds of the total resistance, and for models it may be 90 percent of the total. Wave-making resistance is next in size and becomes increasingly important at high speeds. Eddy-making, such as would occur behind a square sternpost or behind a shaft strut, or, in the case of an extremely full-sterned vessel, behind the entire stern, is small in well-designed vessels. In practice wave-making and eddy-making resistances are treated together as residuary resistance; that is, the resistance remaining when frictional resistance is subtracted from total resistance.

Residuary and frictional resistance do not change with size similarly, but follow different laws. Wm. Froude was the first to realize this, and by so realizing he made ship model testing a success.

Frictional resistance is determined by towing thin planks. In 1888 Froude towed planks up to 50 feet long and showed that frictional resistance can be approximately expressed by the formula

$$R_f = fSV^n$$

where f is a coefficient decreasing with length of surface, S the surface, and n an exponent which for smooth surfaces is a little less than 2. His values for f and n were later revised by his son, R. E. Froude, and still later by a Dutch naval constructor, Tideman. Both Froude's and Tideman's results are given in Reference (b), page 114. Tide-

man's friction is used in connection with "Taylor's Standard Series" (see page 162). Fig. 39 gives the frictional horsepower per 1000 square feet of wetted surface calculated

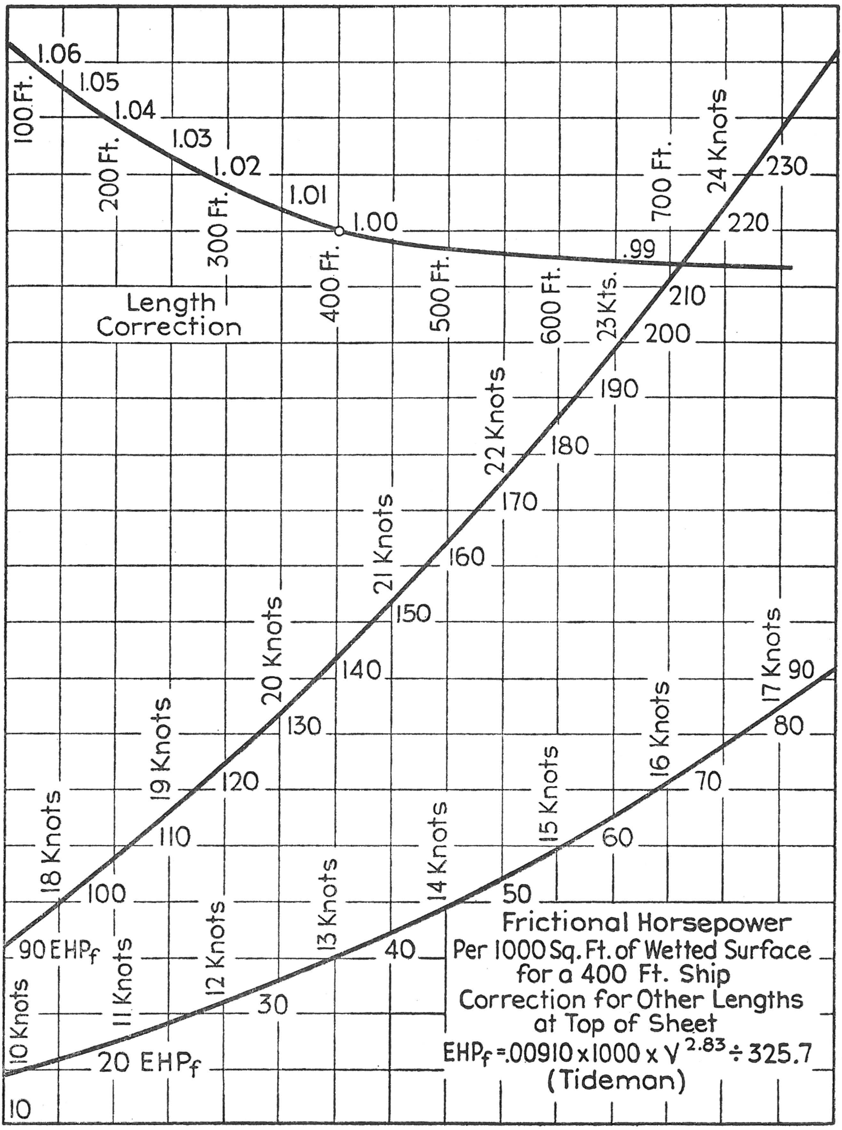


Fig. 39

for a 400-foot vessel and a correction factor for other lengths, using Tideman's values for f and n. Tideman's friction contains an allowance for roughness.

Two other formulas for friction must be mentioned, Gebers and Schoenherr's mean line, both discussed in Reference (*b*). Gebers formula:

$$R_f = \frac{0.0132SV^{1.875}}{L^{0.125}}$$

at 50 degrees F., times a temperature correction factor which amounts to a decrease of about 2 percent for each 10 degrees F. increase in temperature, is used at the U. S. Navy Department's model basin. It is about right for 20-foot models, but is much too low for small models, and it requires a roughness correction for ships. Schoenherr's mean line, so called because it was an average line through a mass of experimental data, is a good approximation for all sizes of models, but it too requires a roughness addition for ships. This formula is complicated and the student is referred to the paper in 1932 *Transactions* of The Society of Naval Architects and Marine Engineers in which it was presented. In this book we will use Tideman's ship friction, Fig. 39.

For frictional resistance, wetted surface of the ship must be calculated. This is done by obtaining the girth of each station up to the waterline, putting through Simpson's rule, and multiplying by a small "obliquity" correction, about ½ percent [see Fig. 13, Reference (*a*)]. Fig. 40 will give a close approximation for normal lines. For a rough approximation, Taylor's formula $WS = 15.6\sqrt{\Delta L}$ is good.

Fouling (a growth of barnacles and weeds) increases frictional resistance greatly. This increase has been estimated to average 5 percent of total resistance for each month since the last cleaning in drydock.

Residuary resistance cannot as yet be expressed by formula, and must be determined by model tests. Mr. Froude proved theoretically that at corresponding speeds (proportional to the square root of the length) residuary

resistance is proportional to displacement. That is, residuary resistance per ton of displacement is constant regardless of the size of the ship. All model tank work is based on this principle. Taylor's "Standard Series" [Reference (*a*)] was

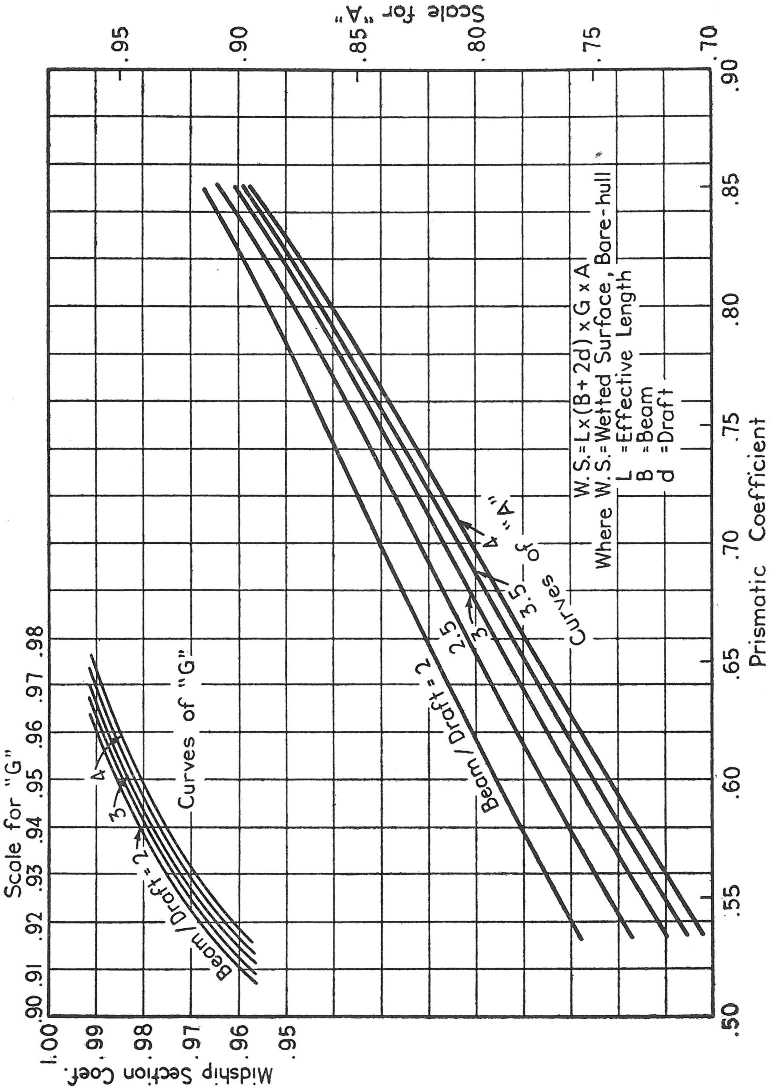


Fig. 40

a comprehensive series of models towed at Washington to establish this "residuary resistance per ton" for all ordinary proportions and coefficients of hulls. For the thirty years since this work was done, the results have been used for estimating power for ships with consistent success.

Estimated Effective Horsepower

There are in general three methods of estimating the power required for a proposed ship:

(a) Taylor's "Standard Series."
(b) Power proportioned from a previous similar ship.
(c) Model tests.

The student need study only the first, because the second requires experience and judgment to be of much use, and when we resort to the third, the tank report gives us the required power. On any important ship the preliminary powering will usually be done by the "Standard Series" and finally be confirmed by a self-propelled model test at Washington.

The results obtained by Taylor from towing his "Standard Series" are embodied in a set of charts contained in his "Speed and Power of Ships," Reference (*a*), giving the residuary resistance in pounds per ton of displacement for a range of speed-length ratio, prismatic coefficient and displacement-length coefficient covering the practical proportions of ships. This is given for two ratios of beam to draft, called B/H ratios by Taylor, and it is assumed that for intermediate ratios of B/H the residual resistance will vary in a straight line between the values found for the given ratios of B/H. Generally, but not always, the wider ship has the higher resistance. The use of the standard series is explained in the text of Reference (*a*), and, since the student must have the book in order to use the charts, only an outline explanation will be given here. A form for the calculation of effective horsepower is shown in Reference (*a*), and the procedure is in outline as follows:

At even values of $V/\sqrt{L}$

(1) Read from the 3.75 B/H charts the proper values of Rr/Δ (Rr/Δ is "residuary resistance in pounds per ton of displacement").

(2) Read from the 2.25 B/H charts the proper values of Rr/Δ.

(3) Interpolate between these for the actual value of B/H.

(4) Read from Fig. 69 the proper values of Rf/Δ.

(5) Correct these for the actual length and wetted surface.

(6) Add corrected Rf/Δ to Rr/Δ, and obtain Rt/Δ; that is, the total resistance in pounds per ton of displacement.

(7) Multiply this by the "*ehp* factor" and the speed-length ratio, or, if preferred, by (displacement $\times$ speed in knots $\div$ 326), obtaining values of *ehp*.

Fig. 69 of Reference (*a*), used for frictional *ehp*, is difficult to read accurately and must be corrected both for actual wetted surface and for length of ship. Table 4, page 180, gives a form for calculating *ehp* using Fig. 39 of this book for ehp_f: ehp_r is calculated just as in Reference (*a*).

Shaft Horsepower

The Standard Series gives the effective horsepower of a bare hull; that is, without appendages such as rudder, struts, bossings, and bilge keels. This bare hull effective horsepower must be increased for the added resistance of the appendages—about 3 percent for a single-screw ship with a rudder the only appendage, about 12 percent for a well-designed twin-screw ship, and perhaps 25 percent for a quadruple-screw ship with struts, bossings and large bilge keels.

The total effective horsepower including appendages is

only part of the shaft horsepower, the remainder being lost in propeller efficiency, etc. The ratio of effective horsepower to shaft horsepower is called the propulsive coefficient, and includes the propeller efficiency, the "wake gain" due to the propeller running in the wake, and the "thrust deduction" due to the suction of the propeller on the hull. These effects are further discussed in Chapter X. The combined effect of these may be a ratio of from 0.60 to 0.75. Accordingly the effective horsepower must be divided by from 0.60 to 0.75 according to the expected propeller efficiency and other effects in order to get the shaft horsepower.

This shaft horsepower is further increased by about 15 percent for "sea conditions"; that is, to allow a margin for wind and waves, a moderate degree of fouling of bottom, etc. If reciprocating machinery is proposed, the shaft horsepower must be divided by a suitable mechanical efficiency factor (see page 108) to get the indicated horsepower.

Increase of Resistance in Shallow Water

This increase is of two distinct kinds. First, as a vessel enters shallow water, there is a gradual increase in resistance beginning when the depth of water is from ½ to ¾ the length of the ship. The resistance steadily increases with increasing shallowness, and the percentage increase is not greatly affected by speed. Second, a sudden and phenomenal increase starts when the speed in knots is about twice the square root of the depth in feet. A speed equal to about 2.5 times this square root is about the limit to which a vessel can be driven, and a speed of 3.3 times the square root of depth is the top of a hump in the resistance curve at which the resistance may be many times the deep water resistance. This is also the speed at which a solitary wave will travel through shallow water. Beyond this speed, the resistance rapidly drops to a value somewhat less than the deep water resistance.

Problems

1. Show that

$$\text{Horsepower} = \frac{\text{pounds resistance} \times \text{speed in feet per minute}}{33{,}000}$$

is equivalent to

$$\text{Horsepower} = \frac{\text{pounds resistance} \times \text{speed in knots}}{325.7}$$

2. If a certain destroyer at 30 knots has just the same resistance as a certain freighter at 15 knots, what is the relation between the effective horsepower of the two vessels at those speeds?

3. What speed of (a) a 16-foot model and of (b) a 400-foot ship "corresponds" to a speed of 20 knots for a 625-foot ship? (c) What speed-length ratio is this for all three?

4. The wetted surface of the example ship up to the 24-foot waterline, by using girths of stations, and an obliquity factor of 1.004 from Reference (*a*), page 16, is found to be 34,690 square feet. What is the resulting wetted surface coefficient C for Taylor's formula $WS = C\sqrt{\Delta L}$?

5. Using Reference (*a*), calculate the Standard Series bare hull effective horsepower for the example ship, at 11,710 tons (24-foot draft) for a range of speed from 12 to 15 knots. The longitudinal coefficient in the standard series charts is the ordinary prismatic coefficient obtained in Problem 1, Chapter II. The wetted surface is given in Problem 4. The displacement-length coefficient is $11{,}710/(4.15)^3 = 164$.

Chapter X

PROPELLERS

References

(*a*) Taylor's "Speed and Power of Ships."
(*b*) "Principles of Naval Architecture," Volume II, Chapter III.
(*c*) Attwood's "Theoretical Naval Architecture," Chapter X.
(*d*) Baker's "Ship Design, Resistance and Screw Propulsion," Volume II.

There are two commonly used means of applying engine power to the propulsion of a ship: screw propellers and paddle wheels. We shall discuss only screw propellers. Discussion of paddle wheels is found in References (*a*) and (*b*).

This verges on marine engineering rather than naval architecture, but the naval architect must work with the marine engineer and should be familiar with the elements of propeller design, and be able to select the proper diameter, pitch and revolutions of a propeller for a given case. We will go only as far as is necessary from that point of view. Entire books, such as Reference (*d*), are devoted to this subject.

The following terms are used in connection with propellers:

Diameter (*d*) is the diameter of the circle traced by the tips of the blades.

Pitch (*p*) is the distance the propeller would advance per revolution, if it were a screw working in a fixed nut. If the driving face is a true screw, or helix, the pitch of that face can be measured on the propeller; this is the "nominal" pitch and is used for design purposes. The "effective" pitch of blades of any shape is the distance the propeller advances per revolution when the thrust is zero.

Pitch ratio is the ratio of pitch to diameter.

Thrust (T) is the axial force exerted by the propeller when driving the ship.

Wake. The forward motion of the water around the stern relative to undisturbed water is from three causes: (1) the frictional wake due to the frictional resistance of the vessel, (2) streamline wake due to the closing in of the water behind the stern, and (3) wave wake due to the orbital motion of the water in the ship's wave pattern. The resultant wake is greatest near the surface and close to the ship. The average wake in way of a centerline propeller may be about 25 percent of the ship's speed, and in way of a twin-screw installation, about 10 percent of the ship's speed.

Wake-fraction (w) is the ratio of the average wake to the speed of the ship. There is a gain in propulsive efficiency from the propeller working in the wake which is called "wake gain," and is equal to $1 \div (1 - w)$.

Speed of advance (Va) of the propeller is its speed through the water in which it is working, and is less than the ship's speed by the amount of the wake.

Thrust Deduction. Part of the thrust seems to be offset by suction on the ship developed by the propeller, so that thrust (T) must be greater than resistance (R). The thrust deduction coefficient (t) is the ratio of the thrust deduction to the total thrust, so that $R = (1 - t)T$. t may be from 10 to 20 percent.

Hull efficiency is the net effect of wake gain and thrust deduction, and is equal to $(1 - t) \div (1 - w)$. This may be a net gain of about 10 percent for single-screw vessels, and is about unity for twin-screw vessels. It may be a loss for quadruple-screw vessels.

Slip is the difference between the actual speed of the wheel and the speed it would have if working in a solid. It is called "real" slip if based on speed of advance of the propeller, and "apparent" slip if based on speed of the ship. Real slip is greater than apparent slip by the amount of the wake. If the wake is greater than the real slip, apparent slip will be negative. Real slip must be positive;

that is, the propeller must impart sternward motion to the water, if it is to exert a forward thrust.

Slip ratio (s) is the ratio between slip and speed. For real slip (s), speed of advance of the propeller is used; for apparent slip (s'), speed of the ship is used.

Mean width ratio is the ratio of mean width of a blade to the diameter of the propeller.

Disk area ratio is the ratio of the developed area of all the blades to the area of the circle whose diameter is that of the propeller. These two area ratios are related, and, if the boss diameter is $0.2D$, an average figure, the disk area ratio equals $0.51 \times$ mean width ratio $\times$ number of blades.

Propeller efficiency (e) is the ratio of power developed by the propeller, expressed as thrust times speed of advance, to power supplied to the propeller, expressed as torque times revolutions. This ratio cannot be made much more than 70 percent. Much of the loss of power is in the energy of the "race" due to its sternward velocity. This can be reduced by imparting a slow motion to a large mass of water rather than a fast motion to a small mass of water. This requires a large propeller for best efficiency. This superiority of large slow-turning propellers is the reason for using reduction gears with fast-turning turbines or engines.

Table 1 is a collection of useful formulas relating to the above terms.

Propeller Design

In addition to the selection of suitable diameter, pitch and revolutions, the design of a propeller includes drawing the details of the propeller, study of its strength, and a check against cavitation. We will take up only the selection of diameter, pitch and revolutions, with a short discussion of cavitation.

Each one of the four references presents a different method of propeller design with the necessary charts. Each method is in certain respects better than the others. We will, however, apply only the method given in Reference (a), and,

TABLE 1.—FORMULAS RELATING TO PROPULSION

Let

V = speed of ship in knots
V_A = speed of advance of propeller through water in knots
p = pitch of propeller in feet
D = diameter of propeller in feet
rpm = revolutions of propeller per minute
N = number of propellers
shp = shaft horsepower
ehp = effective horsepower
R = resistance of towed ship in pounds
T = thrust required to propell ship in pounds
$\mathcal{Q}$ = torque in one shaft in foot-pounds
s = real slip, in per cent of V_A
s_1 = apparent slip, in percent of V
w = wake fraction, in percent of V
t = thrust-deduction factor
e_P = efficiency of propeller

Then

$$shp = \frac{2\pi \times rpm \times \mathcal{Q} \times N}{33{,}000} = \frac{rpm \times \mathcal{Q} \times N}{5252}$$

$$\mathcal{Q} = \frac{33{,}000\ shp}{2\pi \times rpm \times N} = \begin{cases} 5252\ shp/rpm \text{ for single screw} \\ 2626\ shp/rpm \text{ for twin screw} \end{cases}$$

$$V_A = \frac{p \times rpm \times (1 - s)}{101.3} = V(1 - w)$$

$$\frac{V}{V_A} = \frac{1}{(1 - w)} = \text{wake gain} \qquad w = \frac{V - V_A}{V} = 1 - \frac{V_A}{V}$$

$$V = \frac{p \times rpm \times (1 - s)}{101.3 \times (1 - w)} = \frac{V_A}{(1 - w)}$$

$$(1 - s) = (1 - s_1) \times (1 - w), \text{ or } w = 1 - \frac{1 - s}{1 - s_1}$$

$$s = \frac{p \times rpm - 101.3\ V_A}{p \times rpm}$$

$$s_1 = \frac{p \times rpm - 101.3\ V}{p \times rpm} = 1 - \frac{1 - s}{1 - w}$$

$$t = 1 - \frac{R}{T} = \frac{T - R}{T} \qquad \frac{T}{R} = \frac{1}{1 - t}$$

$$R = \frac{ehp \times 325.7}{V} \qquad T = \frac{ehp \times 325.7}{V(1 - t)} = \frac{R}{1 - t}$$

$$e_P = \frac{T \times p \times rpm \times (1 - s)}{2\pi\mathcal{Q} \times rpm} = \frac{T \times V_A \times 101.3}{2\pi\mathcal{Q} \times rpm} = \frac{ehp}{shp} \times \frac{1 - w}{1 - t}$$

in particular, Figs. 204, 208, 211 and 215 of Reference (*a*).

These figures are all based on a basic coefficient B, equal to $\frac{rpm \times \sqrt{\text{power}}}{\text{speed of advance}}$. B is further described by subscripts p and u, according to whether *shp* (called P by Taylor) or *ehp* (called U by Taylor) is used, and 3 or 4, according to whether three-bladed or four-bladed propellers are used. Thus B_{p3} refers to B for a three-bladed propeller, using *shp* in calculating B. Figs. 204 and 211 are based on power delivered to the propeller (*shp*), while Figs. 208 and 215 are based on power delivered by the propeller [$ehp \times (1 - w)/(1 - t)$], for three and four-bladed propellers, respectively.

Number of Blades

Three-bladed propellers are generally slightly more efficient than four-bladed propellers. When partially immersed, as in a lightly loaded cargo ship, a four-bladed wheel may give less vibration. Four-bladed propellers for merchant vessels and three-bladed propellers for naval vessels are usual, but not invariable.

Example. We will choose a propeller for the example ship. In Chapter I this ship is said to have 3300 indicated horsepower for 12 knots. We will assume:

shp = 2700
ehp = 1700 (This includes an allowance for sea power of about 15 percent over that obtained in Problem 4, Chapter IX)
Wake fraction (w) = 0.20
Thrust deduction coefficient (t) = 0.12

Then

$$\text{Useful propeller power} = 1700 \times \left(\frac{1 - 0.20}{1 - 0.12}\right) = 1550$$

$$\text{Speed of advance } 12 \times (1 - w) = 9.6 \text{ knots}$$

In the following, the notation used by Taylor in Reference (*a*) is used; namely,

N is revolutions per minute

δ is "a quantity dealing with size" $= \dfrac{Nd}{V_A}$

d is diameter in feet
p is pitch in feet
P is shaft horsepower

From Fig. 211, Reference (*a*), it is seen that the highest attainable efficiency is about 0.73, with a B_{p4} of about 5 and a δ of about 90. To have $B_{p4} = 5$, since $B = \dfrac{N \times \sqrt{P}}{(V_A)^{2.5}}$ N would have to be $\dfrac{5 \times (9.6)^{2.5}}{\sqrt{2700}} = \dfrac{5 \times 2.86}{52} =$ about 28. Also, with $N = 28$, and $\delta = 90$, d would be $\dfrac{9.6 \times 90}{28} =$ over 30 feet. 28 *rpm* is impossibly slow, and 30 feet diameter is impossibly large. The lowest revolutions at which a reciprocating engine of practicable proportions could develop 3300 indicated horsepower would be about 80. Also the largest diameter that would be feasible on 24-foot draft would be about 19 feet.

Assuming $N = 80$, using Fig. 211, Reference (*a*),

$$Bp4 = \frac{80 \times \sqrt{2700}}{(9.6)^{2.5}} = 14.6$$

Pitch ratio (*a*) for maximum efficiency........	0.95	
Maximum efficiency........................	0.67	
δ..................................	146	
$d = \delta \times 9.6/80$..........................	17.5	feet
$p = 17.5 \times 0.95$........................	16.6	feet

This is about the best that can be done under the assumed conditions. If Fig. 215 is used, with $Pu = 1550$, it will be found that substantially the same results are obtained.

Cavitation

Cavitation occurs when the water is unable to close behind the blade fast enough to prevent cavities. It depends on many factors, including thrust per square inch of developed area, tip speed, shape of section of blade, and depth of immersion. According to Reference (*c*), the thrust should not

exceed about 12 pounds per square inch, although Reference (*a*) cites cases where this criterion is not applicable. Nevertheless, for ordinary merchant vessels it may be used as a guide.

Applying this simple criterion to the case just worked out, assuming a mean width ratio of 0.25, as is the basis of Fig. 211,

$$\text{Developed area} = 4 \times 0.25D \times 0.4D = 122.5 \text{ square feet}$$

$$\text{Thrust} = \frac{326 \times ehp}{V(1 - t)} = \frac{326 \times 1700}{12 \times 0.88} = 52{,}500 \text{ pounds (Fig. 41)}$$

Thrust per square inch = 52,500 ÷ (122.5 × 144) = 3.0 pounds. This is far under the criterion of 12.5 pounds per square inch.

Chapter XI

SAFETY: FREEBOARD, FLOODABLE LENGTH AND STABILITY FLOODED

References

(*a*) Load Line Regulations of the United States (1941).

(*b*) "Principles of Naval Architecture," Volume I, Chapters II and V.

The three subjects of freeboard, floodable length and stability flooded are grouped in one chapter, because each of them, in its own way, tends to prevent the loss of the ship due to the admission of the sea.

Freeboard is the distance from the top of the hull to the water, or, more exactly, the distance from the top of the "uppermost complete deck having permanent means of closing all openings in the weather portion," at the side of the ship amidships, to the waterline. Freeboard tends to keep water out of and off of a ship, and minimum allowable freeboard for all merchant ships is prescribed by law.

Floodable length at any given point of a ship is the length which, if filled with sea water to the level of the sea (the middle of the length being at the given point), will just bring the bulkhead deck at its lowest point to 3 inches above the water. Unless transverse watertight bulkheads are so spaced that water admitted to the ship because of a collision, etc., will be confined to a length not greater than the floodable length, the ship may founder; that is, settle bodily and sink (provided it did not first capsize because of instability). The 3 inches referred to is simply a slight statutory margin.

Stability flooded is the stability of a ship partly filled with water as a result of flooding. Adequate stability

flooded prevents the ship from capsizing due to loss of stability caused by the free surface of the water in the flooded spaces.

Freeboard

Freeboard legislation began in 1876 when Samuel Plimsoll, an English reformer, persuaded Parliament to require a "Plimsoll Mark" (the familiar circle and bar $\ominus$) to be painted on the side of every ship at a height chosen by the owner as a mark beyond which he would not load his ship. For the development of freeboard laws since then, see Reference (*b*), Chapter II. The present rules are the outcome of the "International Load Line Convention of 1930," and are embodied in Reference (*a*), giving freeboard rules for steamers, tankers, sailing ships, and ships carrying deck cargoes of lumber. Only the rules for steamers will be studied here.

Adequate freeboard protects the ship in four ways:

(a) It provides a reasonably long curve of righting levers, as was shown in Chapter III.

(b) It prevents the ship from being overloaded in regard to her structural strength. Chapter VIII showed that the bending moment in the hull girder is approximately proportional to displacement.

(c) It tends to keep the seas from breaking over the weather deck as solid water and staving in the hatches.

(d) It aids the seamen in working on the weather deck in heavy weather.

The required freeboard is based primarily on the length between perpendiculars of the ship, as tabulated in Rule 43.67 of Reference (*a*), and increases with the length. Five corrections are applied to this "table-freeboard," as follows:

Fullness correction. [Rule 43.67 (*c*)]. A full ship must have more freeboard than a finer ship.

Depth correction. [Rule 43.67 (*d*)]. A deep ship must have more freeboard than a shallower ship.

Superstructure correction. (Rule 43.53). A ship with short superstructures (or none) must have more freeboard than one with longer superstructures.

Sheer correction. (Rule 43.54 to 43.59). A ship with little sheer must have more freeboard than one with more sheer.

Camber correction. (Rule 43.60 to 43.61). A ship with little camber must have more freeboard than one with more camber.

These corrections are all simple except the superstructure correction. This is more complicated because the effectiveness of the superstructures, as well as their length, is considered. This effectiveness depends on the tightness of the doors and hatches enclosing the superstructures, all as laid down in Rules 43.46 to 43.52, inclusive, of Reference (*a*).

"Class 2" closing appliances, in Rules 43.44 and 43.45, can be made to comply with the requirements for "tonnage openings" in the rules for "measurement of vessels" (see Chapter IV), and the usual poop, bridge and forecastle can be exempted from tonnage and still be 100 percent effective for freeboard.

If an opening with "temporary closing appliances" (that is, a tonnage opening) is fitted in a deck which would otherwise be the freeboard deck, to exempt space below from tonnage measurement, the deck below becomes the freeboard deck, and the permissible draft is reduced. For instance, a flush-deck vessel with forecastle, 450 feet long and 30 feet deep, might have a freeboard of about 84 inches and a draft of 23 feet. But if a tonnage opening is fitted in the upper deck, reducing the freeboard depth to say 22 feet 6 inches, with complete superstructure, the freeboard would be about 87 inches, minus $22\frac{1}{2}$ inches for L/D ratio, minus 42 inches for superstructure, or, say, 22 inches, giving a draft of about 20 feet 8 inches instead of 23 feet. In this case, unless an especially low-density cargo was intended, the deadweight carrying capacity might be too small for the cargo "cubic" (volume) provided.

Tankers are permitted to have less freeboard than steamers because (a) their deck openings, which are relatively small oiltight hatches, are more secure than cargo hatch covers, (b) they can jettison or dump cargo in an

emergency without opening hatches, and (c) if they do dump oil, breaking of the seas is stopped. They must, however, have a forecastle, whose length is at least 7 percent *L*. As an example of the allowed reduction in freeboard, the basic freeboard for a 400-foot steamer is 71.5 inches, and for a 400-foot tanker 62.5 inches.

Problem in Freeboard

1. Assume that the particulars of the example ship affecting freeboard are as follows:

Length between perpendiculars—420 feet.
Depth—30 feet 3 inches plus ⅝ inch stringer plate.
Block coefficient at 25.71-foot draft = 0.76.
Forecastle 35 feet long, with class 2 doors.
Bridge 120 feet long, with class 1 doors forward, class 2 aft.
Poop 40 feet long, with class 2 doors.
Sheer and camber—Standard.

Show that the designed draft of 24 feet molded complies with Reference (*a*).

Suggestion: Arrange the work as follows (write the proper figures in the spaces provided, indicated by [ ]).

Tabular freeboard for 420 feet *LBP* .... [ ]

Block coefficient correction $\dfrac{[\quad] + 0.68}{1.36} \times [\quad] = [\quad]$

Depth correction $\left([\quad] - \dfrac{LBP}{15}\right) \times 3'' = [\quad]$

Effective length of superstructures (Rules 43.47, 43.49 and 43.50).
Poop [ ] × [ ] percent, Bridge [ ] × [ ] percent, Forecastle [ ] × [ ] percent.
Aggregate effective length = [ ] feet, = [ ] percent of *L*.
Correction (Rule 53) [ ] percent of 42″ .... [ ]

[ ]

No correction for sheer or camber.
Summer freeboard = [ ] = [ ] feet [ ] inches.
Allowable draft = [ ] − [ ] = [ ]

Before studying floodable length and stability flooded, the effect of admitting the sea into a compartment of a ship should be considered. This effect is twofold:

(a) The ship will settle and may change trim. If the bulkhead deck is brought below the level of the sea, water may enter other compartments until the ship sinks. Transverse stability is not involved in this phase.

(b) *GM* will be changed. *VCB* is increased and *BM* decreased. The net change may be a reduction in *GM* greater than the original *GM*, leaving the ship unstable.

Study of floodable length shows the spacing of bulkheads required to prevent disaster due to sinkage and trim; study of stability flooded shows the *GM* required to prevent disaster due to capsizing.

Floodable Length

At present, floodable length regulations [Reference (*a*), Part 46] apply only to passenger vessels, but the principles involved apply equally to all vessels. A typical floodable length curve is shown in Fig. 41. It is a curve

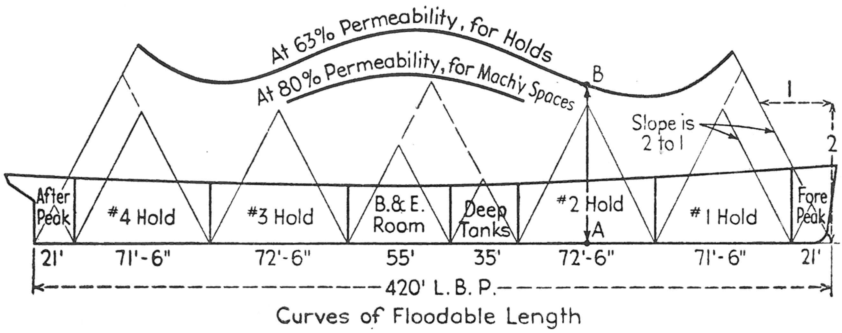


Fig. 41

drawn so that at any point in the ship, such as *A*, the height of the curve above the base line *AB* is the length that can be opened to the sea and flooded (with *A* as the midpoint of the length) and just bring the margin line (3 inches below the top of the bulkhead deck at side) to the level of the sea.

The sketch shows curves for "63 percent permeability" and for "80 percent permeability," and shows that the higher permeability results in a shorter floodable length.

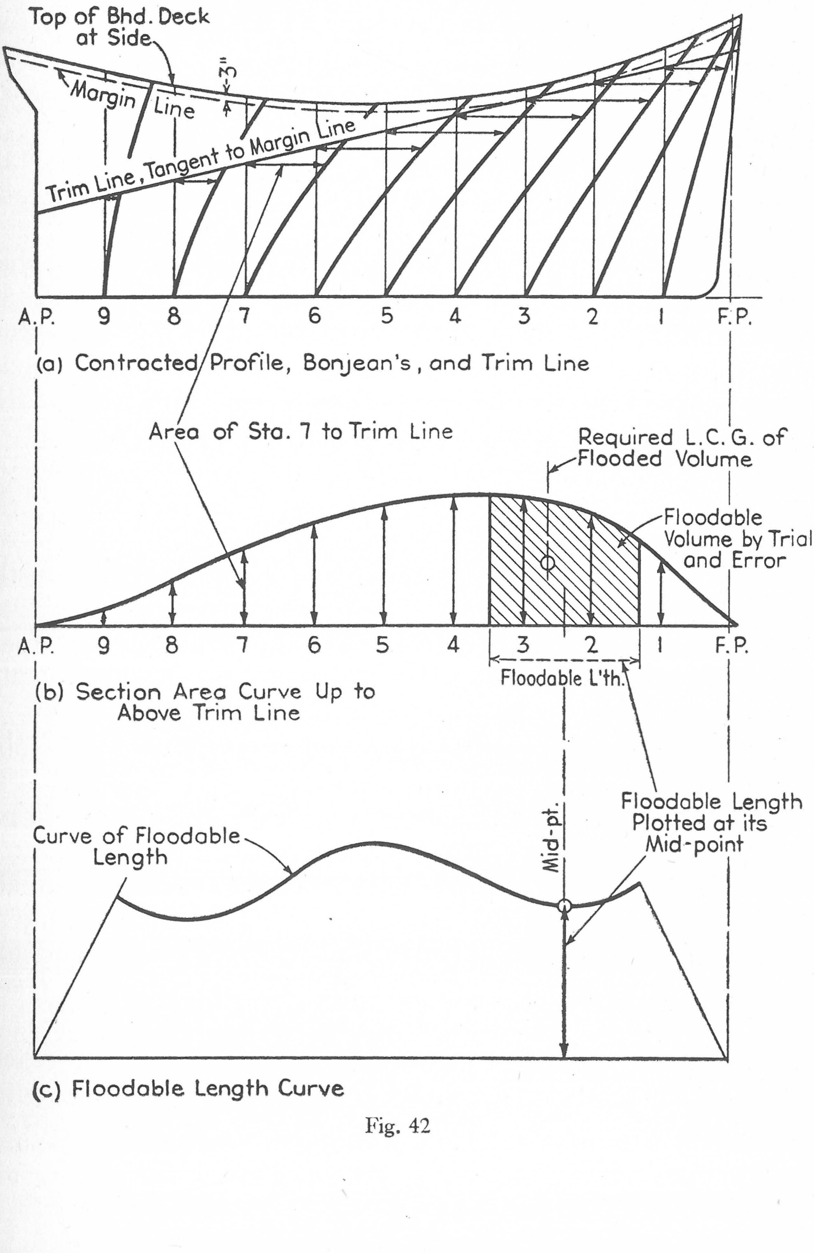


Fig. 42

This is because a high permeability will admit more water in a given length than will a low permeability. The permeabilities of living spaces and of machinery spaces are fairly definite, but the permeabilities of cargo holds may vary widely, even in a given ship from voyage to voyage. So, in practice, permeability is assumed to be given by the formulas in Reference (*a*), Section 46.3. Obviously the resulting floodable length curve may apply only approximately to the ship as actually loaded on a given voyage.

If the floodable length curve is drawn to the same scale vertically as is used horizontally, as in Fig. 41, then any compartment may be tested by drawing lines from the ends of that compartment at a slope of 2 to 1, as shown. If they meet below the floodable length curve, the length of the compartment is less than the floodable length, and flooding the compartment would not immerse the margin line (provided there is still sufficient stability—see third part of this chapter).

The basic method of making a floodable length calculation is outlined below, and illustrated in Fig. 42. Before the calculation can be made, the Bonjean curves must be extended up to the bulkhead deck. Arrangement plans of the ship are required, so that the volumes of living spaces, cargo spaces, and machinery spaces may be obtained, and the assumed average permeabilities calculated by the formulas of Reference (*a*).

The steps involved in obtaining *one point* on a floodable length curve are as follows:

1. The margin line is drawn on the Bonjean curve drawing and a "trim line," or flooded waterline, drawn tangent to the margin line, as in Fig. 42(a). The section area of each station up to the trim line is obtained from the Bonjean curve, and the displacement and *LCB* up to the trim line calculated in the usual manner, using 10 stations. A section area curve is drawn, as in Fig. 42(b).

2. The displacement up to the load line and its moment about *FP*, from the regular form calculations, are subtracted from those up to the trim line, giving the weight of *LCG* of

the "leakage water" which, if admitted to the ship, would sink and trim the ship from the load line to the trim line. This amount of leakage water, divided by the assumed permeability, will give the corresponding floodable volume. For example:

	Tons	Distance from *FP*	Moment
Displacement to trim line	14,200	191.5′	2,719,300
Displacement to load line	11,700	203.6′	2 382,120
Leakage water	2,500	134.9′	337,180

2500 tons × 35 = 87,500 cubic feet.
87,500 ÷ permeability (say, 63 percent) = 139,000 cubic feet, floodable volume

3. Since the leakage water will fill the ship to the level of the sea, that is, to the assumed trim line, this floodable volume will be some part of the section area curve of Fig. 42(b). The question is, what part? Methods have been devised for calculating this, but the trial-and-error method is quicker. For a first trial length, divide the floodable volume by the section area to the trim line in the general vicinity of the required *LCG*. Lay off this trial length with the required *LCG* as its mid-point; read off the forward, middle and after section areas, put them through Simpson's rule (1-4-1 multipliers) and find the resulting volume and the distance from its mid-point to its *LCG*.

4. A much closer second trial can now be made, by correcting the first trial length by the ratio of the required volume to the trial volume, and putting the mid-point as far from the required *LCG* (toward the small end of the section area) as the *LCG* of the first trial volume was from its mid-point. The reason for this will be clear after a study of Fig. 42(b).

5. Similarly, correct the second approximation, until the volume and *LCG* are as required. This gives a point on the floodable length curve as shown in Fig. 42(c).

6. Other trim lines are drawn, and sufficient other points on the floodable length curve are similarly obtained to determine the entire curve of floodable lengths.

This method is sometimes called the direct or the exact method, to distinguish it from various approximate methods, but its exactness is exclusive of the unavoidable and sometimes large error involved in permeability.

F. Schirokauer, in *Shipbuilding and Shipping Record*, September 6, 1928, page 249, devised a simplification of the above procedure, in which seven trim lines are calculated exactly as described above, and intermediate points obtained by a process of interpolation. One trim line is drawn parallel to the waterline, and three are trimmed each way, as in Fig. 43. He found that for ships with normal sheer, these trim lines will give spots well spaced over the floodable length curve.

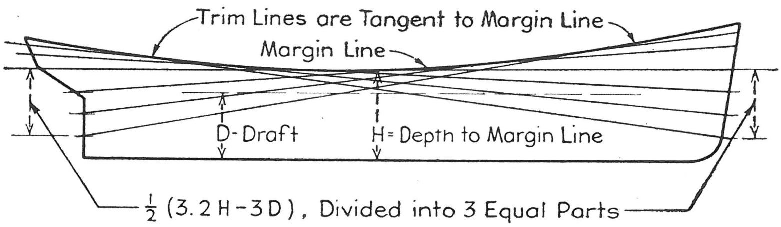


Fig. 43

The British Board of Trade, in its "Instructions to Surveyors," gives an approximate method devised to fit the average results obtained by direct calculations. However, it is no quicker than the direct method as simplified by Mr. Schirokauer.

George Webster (*Transactions* of the Institution of Naval Architects, 1920, or *Marine Engineering*, September, 1920, page 726) has devised an approximation to the Board of Trade method, which, after one has familiarized himself with it, will give an approximate floodable length curve in a few hours. Only a sketch inboard profile and the block coefficient are required. Permeabilities are obtained from the

profile by formulas similar to those of Reference (*a*), but using profile areas instead of volumes. This method is particularly useful for proposed ships, and the student is referred to it for details. The slide rule which Mr. Webster proposed is not necessary and does not expedite the work much.

Permissible length is that part of the floodable length which may legally be used as a compartment length for passenger ships. It is obtained by multiplying the floodable length by a factor of subdivision which varies from a maximum of 1.00 to a minimum of about 0.3, decreasing as the length of the ship increases, and as the character of the ship varies from primarily cargo to primarily passenger. The factor is covered in Section 46.4 (*d*) of Reference (*c*), and in Chapter 5 of Reference (*b*).

It is usual to speak of the greatest number of compartments which is everywhere within the floodable length as a one-compartment, two-compartment, etc., ship, meaning a ship in which any one compartment, any two adjacent compartments, etc., are within the floodable length.

Problem 2

Assume that the floodable length curve of the example ship is as in Fig. 41 and that the length of compartments is as listed.

(a) Are these compartments within the floodable length?

(b) Judging only from the floodable length curve, could the ship survive a collision squarely on the bulkhead between Nos. 1 and 2 holds?

(c) Is there any bulkhead on which the ship could be struck, thus flooding two compartments, without flooding more than the floodable length?

(d) Is this a one-compartment or two-compartment ship?

Stability in Flooded Condition

A ship may have any number of compartments within the floodable length, and yet become unstable and capsize after a collision because of the loss of moment of inertia of the waterplane in the flooded spaces.

In general, the effect of flooding a part of a ship on its stability may be described as follows (see Fig. 44):

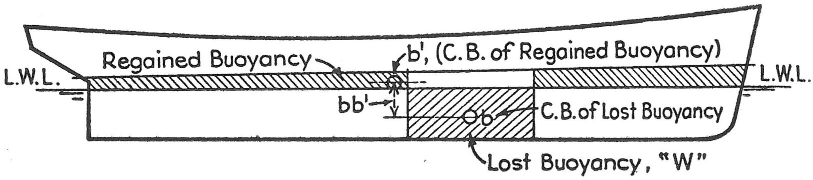


Fig. 44

1. The flooded part of the ship no longer furnishes buoyancy, and the displacement up to the original waterline is therefore no longer equal to the weight of the ship.
2. The ship will settle until the lost buoyancy is regained by a layer (not necessarily parallel) of new buoyancy. This layer will not include any new buoyancy in way of the flooded spaces, because these spaces will fill with sea water as fast as the ship settles.
3. The flooded mean draft will be the original mean draft plus (lost buoyancy ÷ tons-per-inch of remaining waterplane).
4. The *VCB* will be increased by the transfer of buoyancy from the lost spaces up to the layer of new buoyancy, by an amount equal to

$$bb' \times w/\Delta$$

5. The *BM* will be decreased by the loss of part of the moment of inertia of the original waterplane by an amount equal to

$$s \times i/V$$

where

s = permeability of lost waterplane
i = transverse moment of inertia of lost waterplane about the *CG* of the intact waterplane
V = volume of original (and final) displacement

6. The final GM will be the original GM plus the rise of the VCB, and minus the decrease in BM; that is,

$$\text{Flooded } GM = \text{original } GM + bb' \times w/\Delta - si/V$$

The trim resulting from the flooding can be found approximately by calculating the flooded "moment to trim one inch" (see Chapter II). To do this we must first find the flooded longitudinal BM, and to get this BM we must get the longitudinal moment of inertia of the remaining waterplane by subtracting the longitudinal inertia of the lost waterplane from the longitudinal inertia of the whole waterplane, and correcting the remainder to the LCG of the remaining waterplane.

This is somewhat laborious, and is only approximate because the moment to trim one inch changes both with sinkage and with trim. So unless the trim will obviously be negligible (the CB of the flooded spaces near the center of flotation of the original waterplane), it is actually quicker, as well as more accurate, to determine the trim line by trial

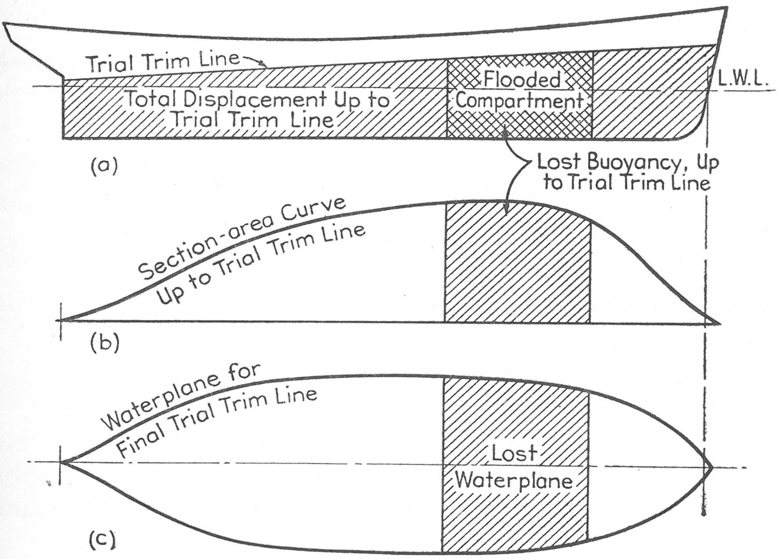


Fig. 45

and error, and then calculate the flooded *GM* directly, as outlined below (see Fig. 45):

1. Choose a trial trim line and calculate the entire displacement and *LCB* up to this trial trim line, as described on page 28.
2. Draw the section area curve, Fig. 45(b), up to this trim line, in way of the flooded spaces, and calculate the amount of the lost buoyancy and its *LCB*, in the flooded spaces.
3. Deduct from this lost buoyancy any intact buoyancy, that is, any tanks, etc., which will exclude the sea even though they are within the flooded spaces, and multiply the remainder by an appropriate permeability factor, obtaining the net lost buoyancy.
4. Subtract the net lost buoyancy from the total displacement up to the trial trim line; the resulting intact displacement and *LCB* should equal the weight and *LCG* of the ship.
5. Move the trial trim line as indicated by the result, and repeat until the required agreement is obtained. This may require four or five trials.
6. Calculate the transverse moment of inertia of the final trial waterplane, Fig. 45(c), and (using a suitable permeability factor) of that part of the waterplane within the flooded spaces. Subtracting the lost inertia from the total inertia gives the inertia of the remaining waterplane.
7. Divide this remaining inertia by the volume of the original (and final) displacement, and get the *BM*.
8. The *VCB* of each station can be obtained by use of an integrator, or can be closely approximated as:

$$\text{(draft at that station)} \times \frac{\text{half-breadth}}{\text{half-breadth} + (\text{half-area/draft})}$$

and the vertical moments of the stations put through Simpson's rule to get the *VCB* up to the final trial waterline. The *VCB* of the lost buoyancy can be similarly obtained, and so the *VCB* of the intact displacement.

9. Flooded *KM* = flooded *VCB* + flooded *BM*.

10. Flooded GM = flooded $KM - VCG$.

It will be found that any submerged intact buoyancy, while reducing the sinkage, will increase the loss of GM by reducing the rise of VCB.

Required *GM* Flooded

If the flooded spaces are symmetrical, so that there is no heeling moment from buoyancy of unsymmetrical intact spaces, GM flooded should be at least positive.

If there is a heeling moment, a positive GM may not be enough to prevent the vessel heeling so as to immerse the margin line and so perhaps flood additional compartments, or from heeling to such an angle as to become unworkable. The latter is about 20 degrees.

Curves of righting levers, such as shown in Fig. 13, are seldom calculated for flooded ships. An approximate curve is needed, however, in order to find the angle of heel. This angle of heel is at the intersection of a curve of heeling lever and a curve of righting lever, drawn as in Fig. 46. The heeling moment of an unsymmetrical intact space is

$$\frac{v\,\mu\,d}{35}$$

and its heeling lever at an angle of inclination of θ is

$$\frac{v\mu d\cos\theta}{35}$$

where

v = volume of intact compartment
μ = permeability of opposite flooded space
d = distance of CG of flooded space from centerline
θ = angle of inclination
Δ = displacement in tons

For small heeling moments, such that the heel is not over 10 degrees, we may use the formula

$$\tan\theta = \frac{v\mu d}{35GM\Delta} \text{ (see page 39)}$$

For larger heeling moments, an approximate expression for righting lever is given by the equation $GZ = GM \sin \theta + K\ BM$ [from Reference (*b*), page 192], where GM and BM are for the flooded ship, and K is as follows:

θ	5°	10°	15°	20°	25°	30°
K	0.00033	0.00258	0.00836	0.01863	0.03320	0.05003

The angle at which the curves of heeling lever and righting intersect, as in Fig. 46, is the angle at which the ship will

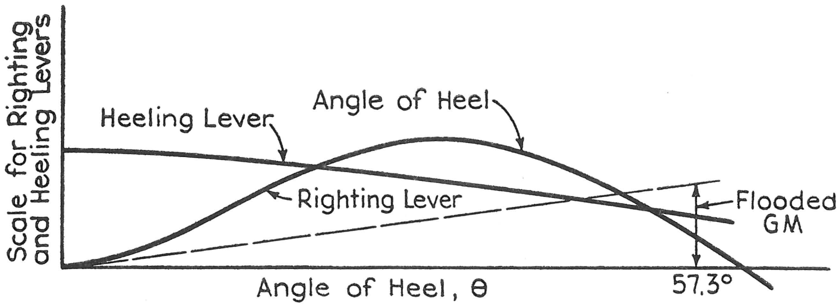


Fig. 46

lie. If this angle immerses the margin line at any point in the ship's length, or exceeds about 20 degrees, it should be reduced either by increasing the original GM or by rearranging bulkheads so as to reduce the heeling moment. A full discussion of this is given in Reference (*b*).

Problem 3

Assume that the example ship, whose subdivision is shown in Fig. 41, is struck on the bulkhead forward of the machinery space while operating at a draft of 20 feet, and with a GM of 3.5 feet.

Assume also that the 3-foot 6-inch inner bottom remains intact, that the vessel is parallel-sided within the two flooded compartments, that trim is negligible, that half of the fuel-oil deep tanks are full, that the tanks from the center-

line to the undamaged side are intact, with a volume of 15,000 cubic feet, and that the top of the fuel-oil deep tanks is 20 feet above the base line.

Form data at the 20-foot draft are given in Problem 2, Chapter II.

What is (a) the *GM* flooded, and (b) the approximate angle of heel due to the intact tanks? For permeabilities use 80 percent for the machinery space, 50 percent for the deep tanks, and 95 percent for the lost waterplane.

Chapter XII

LAUNCHING CALCULATIONS

REFERENCES

(*a*) "Principles of Naval Architecture," Volume I, Chapter VII.
(*b*) Attwood's "Theoretical Naval Architecture," Chapter XIV.

The preceding chapters deal with matters involved in the design of a ship. Launching has nothing to do with design, but since certain aspects of launching are customarily left to the naval architect, the study of theoretical naval architecture must include a discussion of these features of launching.

Fig. 47 shows in barest outline the usual launching arrangements, and illustrates some of the terms which will be used.

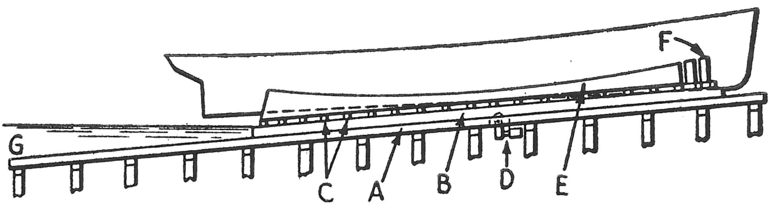


Fig. 47

A are the groundways, resting on a pile foundation. The top of the groundways is covered with a layer of launching grease.

B are the sliding ways, which slide on the groundways and carry the ship into the water.

C are the wedges which are set up hard just before the launching, to prevent the ship from settling when the keel blocks and shores are removed.

D are the hydraulic triggers, one in each way, which hold

the ship from sliding after all keel blocks and shores are out, and which, when all is ready, release the ship. In some yards the ship is released by sawing a timber, or burning a plate, at the head of each sliding way.

E is the packing, which is built up to fill the space between the sliding ways and the bottom of the ship.

F is the fore poppet, which takes the poppet pressure.

G are the way ends, which at a certain stage of the launching exert a severe pressure on the bottom of the ship.

We will review briefly the sequence of events as the ship moves down the ways.

1. As long as the entire length of sliding ways is still on the groundways, nothing of interest happens. As the stern becomes immersed, there is a gradually increasing displacement, which remains small during this stage.

2. As the ship slides out until the stern is overhanging the groundways, the situation shown in Fig. 48(a) arises. There is a tendency for the stern to tip down over the way ends, which is offset by the support of the rapidly increasing displacement of the after part of the ship. This tipping

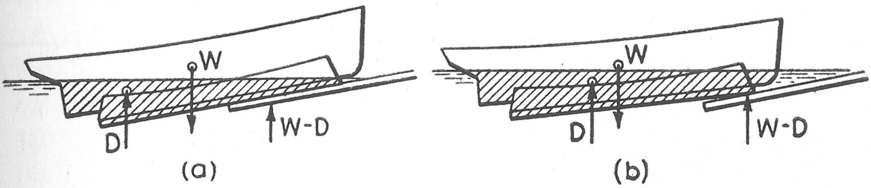


Fig. 48

tendency causes a concentration of pressure on the after end of the ways, called way-end pressure.

As the ship continues to enter the water, and the displacement continues to increase, there comes a time when the displacement is sufficient to lift the stern, Fig. 48(b). This concentrates all the remaining load on the ways on the extreme forward end of the sliding ways; that is, on the fore poppets. This is called pivoting, and the pressure on

the fore poppet at the instant of pivoting is the poppet pressure.

After pivoting, the ship slides along supported by the displacement and the fore poppet until either the fore poppet comes to the ends of the groundways and drops off, or, if the groundways are long enough, it simply floats off without any drop-off. At this point the ship loses the transverse support of the groundways and becomes for the first time dependent on her own stability to remain upright.

The above description suggests four primary requirements of a successful launching:

(a) The ship must start.

(b) There must not be enough concentration of pressure on the way-ends to damage the bottom of the ship.

(c) The fore poppet structure must be adequate to carry the heavy load thrown upon it when the ship pivots.

(d) The ship must be stable when afloat.

Failure to comply with (a) or (b) would be serious; failure to comply with (c) or (d) might be disastrous.

The first thing to be done, before any of these questions can be investigated, is to estimate the probable weight of the ship in its launching condition, including *LCG* and *VCG*. This is done by selecting from the design weight estimate such items as the production schedule indicates will be worked into the ship when launched, listing these with their centers of gravity, and summarizing. It must include the ways and packing, and an allowance for men and dunnage (staging, tools, etc.).

If a launching weight must be estimated before a production schedule is available, we must refer to launching conditions of similar ships. These will be found to vary widely. If the shipway is urgently needed for other work, the ship will be launched as soon as she is structurally fit to stand the stresses. On the other hand, small ships have been launched complete, with steam up, and blowing their own whistle. Nevertheless, usually, launching conditions are somewhat similar. The main steel hull will be practically complete, the superstructures will be largely complete, and

the steel houses may be from 25 to 50 percent complete. Crane clearance may limit the completion of the houses. Most of the hull and deck fittings such as ports, doors, hatches, bitts and chocks, etc., but not the rails, davits, etc., will probably be on board. Carpenter work may consist of practically all of the hold ceiling and sparring, and carpenter decking may be started. The joiner work may vary from nothing to completion; nothing of general application can be said. Cementing and bitumastic coverings may be complete; and about 75 percent of the "paint on steel only" would be a good guess. No rigging or canvas, insulation, equipment or outfit will usually be on board. Machinery weight may consist of the bare boilers, half of their mountings and piping, the condensers and main pumps, the shafting, and usually the propellers.

Next, we should prepare a sketch showing for convenient reference the following data (see Fig. 49). The slope of the ways is so small that horizontal distances may be used for distances along the ways, and vice versa, without appreciable error.

A.* Declivity or slope of ways, in inches per foot.
B.* Declivity of keel.
C.* Expected height of water over way-ends.
D.* Height of keel above groundways at *FP*.
E.* Height of keel above groundways at *AP*.
F.* Distance from way-end to *FP* before start.
G. Distance from assumed center of fore poppet to *FP*.
H.* Distance from forward end of sliding ways to *FP*.
I.* Length of sliding ways.
J. Travel before base line at *AP* reaches water.
K. *LCG* from *FP*.
W. (Not shown) Width of sliding ways.

The first requirement, "the ship must start," is usually

* Dimensions thus designated are usually set by the yard, subject to comment by the naval architect after consideration of their effect in the launching calculations.

a responsibility assumed by the yard, and it is discussed here only as a matter of interest.

Starting depends primarily on the weight per square foot on the grease and the declivity of the ways, and secondarily upon the temperature and kinds of grease used. The weight

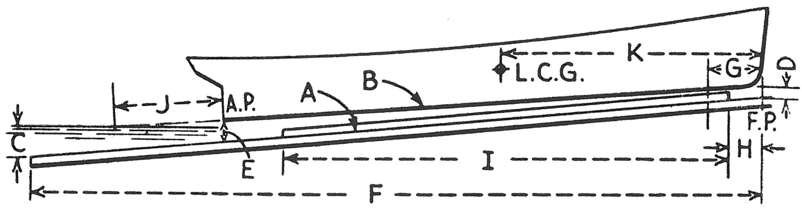


Fig. 49

per square foot of grease is found by dividing the total launching weight, including sliding ways and packing, by the effective area of the sliding ways. If possible, a width of the sliding ways is used which will result in a pressure of from 2 to 2.5 tons, and the declivity of the ways set at approximately $\frac{5}{8}$ inch per foot. At pressures higher than 2.5 tons the grease is liable to burn or squeeze out. For small ships, such as destroyers, yachts, etc., it is impractical to obtain pressures as high as 2 tons. Since the tendency to slide decreases as the pressure decreases, a steeper declivity is required for low pressures. For average conditions, the formula: Declivity in sixteenths of an inch per foot = 18 − (4 × pressure in tons per square foot) will give a reasonable declivity for pressures from 1.0 to 2.5 tons. In any case, the variety, proportions and application of the various greases used are determined by the yard to suit the conditions, including the expected temperature.

Before the next two requirements (relating to way-end pressure and poppet pressure) can be investigated, two calculations must be made—a launching displacement, and calculation of launching curves. A suggested arrangement of the launching displacement calculation is shown in Table 2. Inclined waterlines are drawn on the Bonjean curves

through the base line at stations 12, 8, 4 and *FP*, as shown in Fig. 50. The areas of each station at each inclined waterline are read from the Bonjean curves and summarized as

TABLE 2.—SUGGESTED FORM FOR LAUNCHING-DISPLACEMENT CALCULATION

Station Entering Water		12		8		4		*FP*	
Station	Lever from *FP*	Area	Mom.*	Area	Mom.	Area	Mom.	Area	Mom.
0	0							0	0
1	1							....	....
2	2							....	....
3	3							....	....
4	4					0	0	....	....
5	5					....	....	....	....
6	6					....	....	....	....
7	7					....	....	....	....
8	8			0	0	....	....	....	....
9	9			....	....	....	....	....	....
10	10			....	....	....	....	....	....
11	11			....	....	....	....	....	....
12	12	0	0	....	....	....	....	....	....
13	13	....	....	....	....	....	....	....	....
14	14	....	....	....	....	....	....	....	....
15	15	....	....	....	....	....	....	....	....
16	16	....	....	....	....	....	....	....	....
17	17	....	....	....	....	....	....	....	....
18	18	....	....	....	....	....	....	....	....
19	19	....	....	....	....	....	....	....	....
20	20	0	0	0	0	0	0	0	0
Totals		Sum_1	Sum_2	Sum_1	Sum_2	Sum_1	Sum_2	Sum_1	Sum_2

$$\text{Displacement} = Sum_1 \times LI \div 35 \text{ (if salt water).}$$
$$LCB \text{ from } FP = LI \times Sum_2/Sum_1.$$

* Moment = area times lever from *FP*.

shown in Table 2. Note that the trapezoidal rule is used instead of Simpson's rule. This is not necessary, but it is

more convenient and sufficiently accurate. In the trapezoidal rule, all the quantities are added directly, except the end quantities, half of each of which is included. In our case, the end half-areas are zero, so the half-areas are simply added, as shown. The multiplier, instead of being one-third the longitudinal interval as in Simpson's rule, is the longitudinal interval itself.

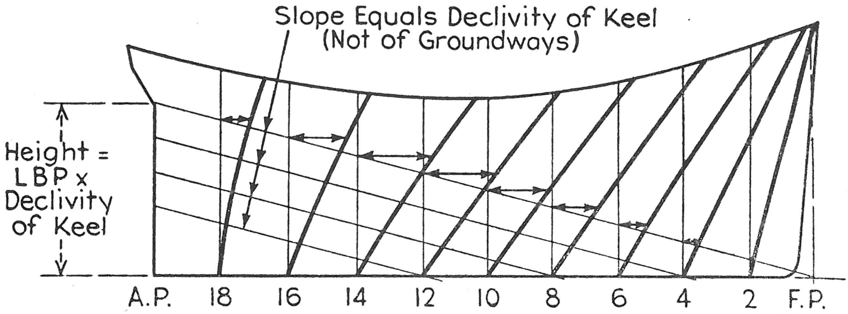


Fig. 50

As regards accuracy, it should be pointed out that the entire launching calculation is based on an assumed infinitely slow launch and a level water surface, while, in fact, the entrance of the ship into the water causes large deformations in the original surface of the water, and the dynamic effect of the relative movement of ship and water (particularly as tending to lift the stern) is considerable. As a result of ignoring these effects, the launching calculations cannot have the degree of accuracy which can be obtained in, for example, curves of form calculations.

To the molded displacement just obtained must be added an approximate displacement of the sliding ways and packing. The total displacement of ways and packing can be calculated approximately, and a displacement per linear foot used. An allowance for appendages should be included, but painstaking accuracy in determining just how much of these is immersed is not justified. The sum of molded displacement, appendages and ways-and-packing

gives the total displacement (and *LCB*) at each of the waterlines chosen.

When the displacement calculations are finished, the real launching calculations are started. A suggested arrangement is shown in Table 3. The use of the various items will be explained on the following pages. The capital letters used in Table 3 refer to corresponding items in Fig. 49. This

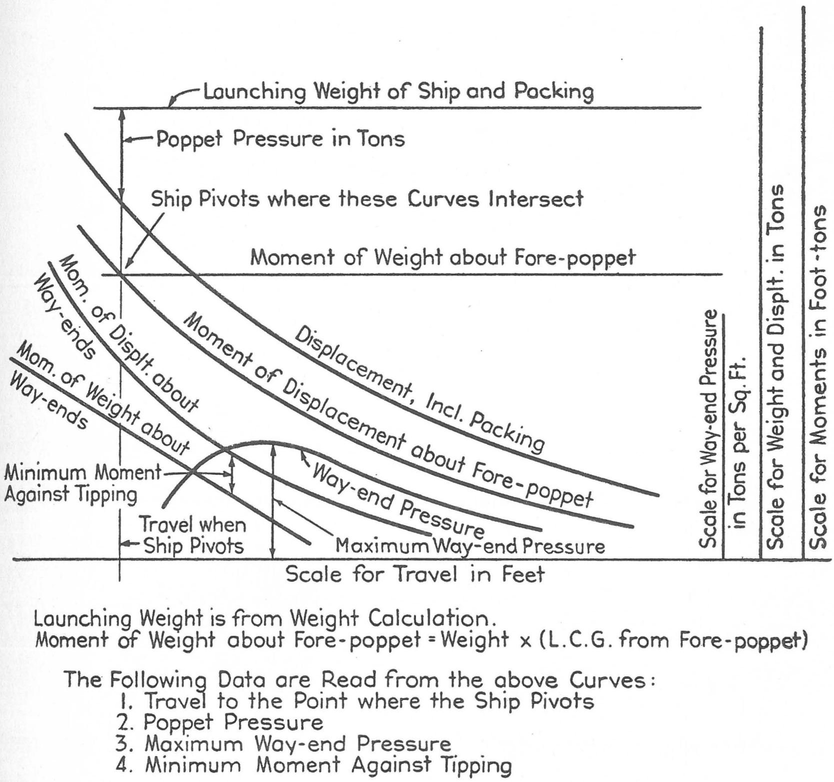


Fig. 51

calculation is for the purpose of obtaining the items marked with an asterisk, which are plotted on a base of "travel" as "launching curves." Fig. 51 shows typical launching curves—these are illustrative only, and are not to scale.

Way-End Pressure. (See Fig. 52.) Numbers in circles

TABLE 3.—ARRANGEMENT OF LAUNCHING CALCULATION

(Figures refer to Fig. 52)

				12	8	etc.
(1)	Station entering water			12	8	etc.
(2)	Travel		$= J + [20 - (1)] \times$ L. I.	..	..	..
(3)*	Displacement		(from Table 2)	..	..	..
(4)	*LCB* from *FP*		(from Table 2)	..	..	..
(5)	*LCB* from poppet		$= (4) - G$	..	..	..
(6)*	Moment of displacement about poppet		$= (3) \times (5)$	..	..	..
(7)	*LCB* from way-end		$= F - [(4) + (2)]$	..	..	..
(8)*	Moment of displacement about way-end		$= (3) \times (7)$	..	..	..
(9)	*LCG* from way-end		$= F - [K + (2)]$	..	..	..
(10)*	Moment of weight about way-end		$=$ Wt. $\times (9)$	..	..	..
(11)	Moment against tipping		$= (8) - (10)$	..	..	..
(12)	Load on ways		$=$ Wt. $- (3)$	..	..	..
(13)	Center of load on ways from way-end		$= (11) \div (12)$	..	..	..
(14)	Length of ways in contact		$= F - [H + (2)]$	..	..	..
(15)	Ratio of (13) to (14)		$= (13) \div (14)$	..	..	..
(16)	Mean pressure on grease		$= (12) \div [(14) \times 2w]$	..	..	..
(17)	If (15) is between 0.333 and 0.667	Maximum pressure factor	$= 4 - [6 \times (15)]$	..	..	..
(18)*		Way-end pressure	$= (16) \times (17)$	..	..	..
(19)	If (15) is less than 0.333	Maximum pressure factor	$= 2 \div [3 \times (15)]$	..	..	..
(20)*		Way-end pressure	$= (16) \times (19)$	..	..	..
(21)	Moment of weight about fore poppet		$=$ Wt $\times (K - G)$	[	]	
	Travel to pivoting, from curves			[	]	
	Poppet pressure, from curves			[	]	
	Minimum moment against tipping, from curves			[	]	
	Maximum way-end pressure, from curves			[	]	

* These items are plotted on a base of travel as "Launching Curves," as in Fig. 51.

refer to items in Table 3. The moment of the weight of the ship, tending to tip the stern down over the way-ends, is $[W \times (9)]$. This must be more than offset by the upward moment of the displacement $[(3) \times (7)]$. The excess of the latter over the former is the "moment against tipping," item (11). This moment against tipping must be balanced by an opposing moment of the pressure still exerted by the ways, item (12). This pressure is the difference between the weight of the ship and the displacement, so the amount of (12) must be $[W - (3)]$. Its moment

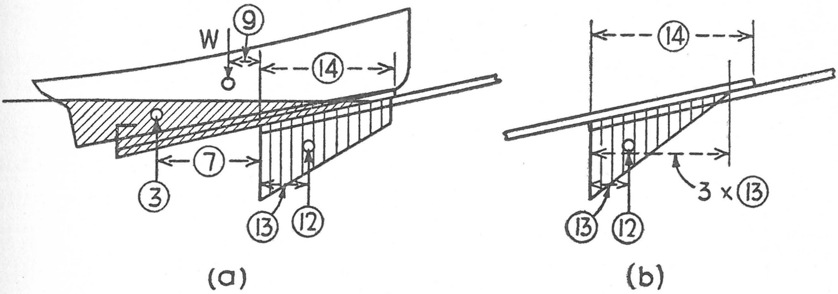


Fig. 52

about the way-ends must equal the moment against tipping, item (11), so its lever about the way-ends must equal (11) divided by (12). This gives item (13). If (13) is not exactly in the middle of the length of ways still in contact, item (14), the distribution of (12) cannot be uniform; it must be greater at the end which (13) is nearest. We do not know exactly how this load will be distributed, for the distribution will depend on the amount of yielding of the ways and deflection of the ship. So for simplicity we assume that the pressure is distributed varying in a straight line as shown in Fig. 52. If (13) is between ⅓ and ⅔ of (14) the pressure is assumed to be distributed as a trapezoid, as in Fig. 52(a), and the pressure at the after end can be proved to be

$$P \text{ maximum} = P \text{ average} \times (4 - [16 \times (15)])$$

If (13) is less than ⅓ of (14), the pressure diagram is assumed to be a triangle as in Fig. 52(b), and the pressure at the after end can be proved to be

$$P \text{ maximum} = P \text{ average} \times \frac{2}{3 \times (15)}$$

If (13) is more than ⅔ of (14), there is no pressure at the way-ends, the diagram is like the Fig. 52(b) turned end for end, and the heavy pressure comes on the forward end of the sliding ways; that is, on the fore poppet. Since this is designed to take the entire poppet pressure a moment later, as we shall see below, we are not particularly interested in the case where (13) is more than ⅔ of (14).

Allowable way-end pressures depend on the strength of the bottom of the ship to take the pressure.

For merchant ships pressures of 10 to 12 tons per square foot are not unusual, but naval ships, with, usually, 4-foot frame spacing, cannot usually take more than 6 or 7 tons successfully. However, for naval ships particularly, this is entirely a matter of the internal arrangements of the ship, such as location of bulkheads, riveting and welding details, etc. For a heavy ship this subject calls for careful analysis of the strength of the bottom of the ship in way of maximum way-end pressure. Bottom damage has occurred on battleships with a pressure of 10 tons per square foot. On the other hand, the *Saratoga* (*Transactions* of the Society of Naval Architects and Marine Engineers, 1925) successfully took a calculated pressure of 15.1 tons per square foot, resulting from an unexpectedly low tide.

If the way-end pressure is excessive, it can be reduced by

(a) adding water ballast forward,
(b) extending the ways further into the water,
(c) increasing the declivity of the keel.

Poppet Pressure. The downward moment of the weight about the fore poppet [line (21), Table 3] is constant throughout the launching. But the upward moment of the displacement about the fore poppet [line (6), Table 3] is steadily

increasing as the ship enters the water. When these become equal, the displacement will lift the stern, the ship pivots about the fore poppet, and the entire pressure on the ways is taken on the poppet. The pressure on the ways is always the weight minus the displacement. So in Fig. 51 directly over the intersection of these two moment curves, the distance between the weight and displacement curves will give the poppet pressure. The poppet is designed to take this load by using the principles of "Strength of Materials."

There are three forces acting at the head of a poppet, as illustrated by Fig. 53; namely, the tension in the saddle plate T, the compression in the poppet R, and the normal pressure against the shell C. If T is drawn parallel to the

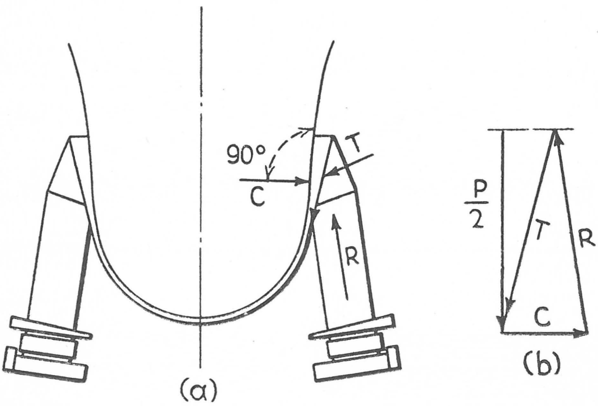


Fig. 53

upper part of the saddle plate to scale so that its vertical component equals $\frac{1}{2}P$, and lines are drawn from the end of T parallel to the poppet and perpendicular to the shell, as in Fig. 53(b), a triangle of forces is obtained from which the values of C and R can be scaled.

If T is large, several saddle plates may be required. Then some means of avoiding concentrating the poppet pressure on the forward edge of the forward plate as the ship pivots

is required. Two such means are shown in Fig. 54. The spacing of the crushing strips in Fig. 54(a) may be calculated so as to give uniform maximum loading on each saddle plate as the ship pivots.

In a small vessel the poppets may be simply headed against an angle clip welded to the shell plating, and no saddle plate used.

There remains the fourth requirement—stability when afloat. To find *GM* we must know *G* and *M*. The weight calculation has given us *VCG*, so we know *G*. *M* is obtained from the curves of form, but two corrections are required. The height of *M* is the sum of *VCB* and *BM*. *VCB* must be corrected for the displacement of the ways and packing, which are much lower than the rest of the displacement. Also *BM*, which is equal to I/V, must be corrected because the ways and packing increase V, but do not increase I, and so the net effect is to reduce *BM*. The combined effect may be to lower *M* by a foot or more, so these corrections cannot be neglected. The following typical figures show how this may be arranged:

Launching weight, including ways..........4000 tons—21.5 feet *VCG*
Displacement of ways........ 300 tons, $VCB = 0.0$
Displacement of ship.........3700 tons, $VCB = 5.6$ feet (from curves of form)

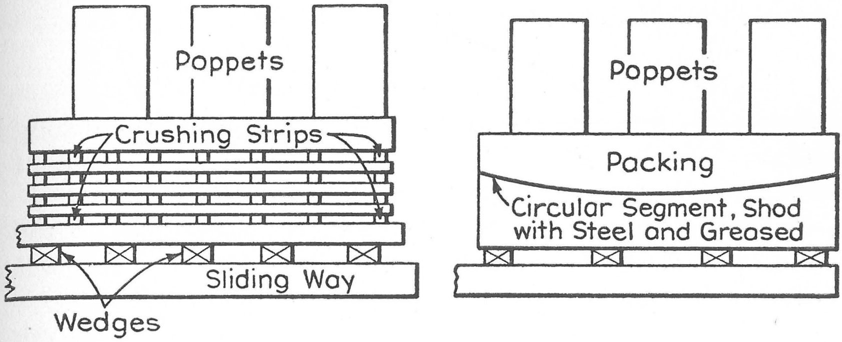
Displacement including ways..4000 tons, $VCB = 5.2$ feet

$BM = (KM - VCB) = (28.7 - 5.6) = 23.1$ feet (from curves of form)
Corrected $BM = 23.1 \times 3700/4000 = 21.4$ feet
Corrected $KM = 21.4 + 5.2 = 26.6$ feet
$GM = 26.6 - 21.5 = 5.1$ feet

This *GM* will be increased somewhat by the trim by the stern which the ship will probably have in the launching condition. This increase is usually neglected, as a margin of safety. If *GM* is found to be very small, or negative, water ballast will be added in inner-bottom tanks as necessary to lower the *VCG* until *GM* is 2 feet or more.

Static Drop. If the depth of water over the way-ends is

less than the draft to the bottom of the sliding ways when the ship is afloat, the difference between the two is called the static drop, and the ship drops off of the end of the ways with a noticeable "curtsy".



(a) (b)

Fig. 54

Dredging. To determine whether dredging is required, draw a profile showing the ways, the location of the stern when the ship pivots, the location of the stern when the ship drops off or floats off, and the bottom of the river, from soundings, as shown in Fig. 55. The "probable path" is drawn wholly by eye, somewhat below the straight line

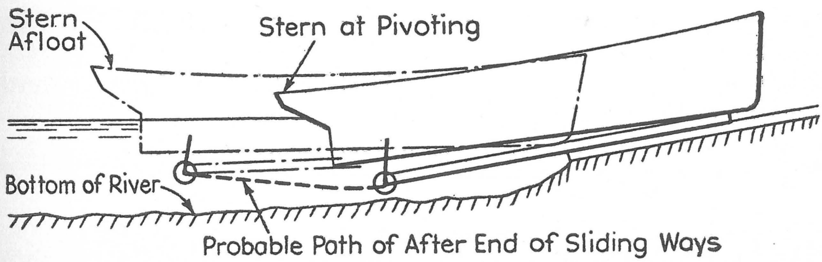


Fig. 55

between the two locations of the bottom of the after end of the sliding ways. There should be a clearance of two feet or more.

Internal Shoring. Sometimes the ship must be strengthened by temporary internal shoring in way of the fore poppets, or of the ways in way of the maximum way-end pressure. Each such case is a separate problem, and the student should notice what is being done in any launchings in which he has the opportunity to do so.

Main Hull Stresses during Launching

These must sometimes be investigated in the case of long shallow vessels, or vessels launched at an early stage of completion. There are three cases to investigate:

1. A hogging stress at the travel corresponding to the "minimum moment against tipping."
2. A sagging stress when the ship pivots.
3. The shear directly over the fore poppet when the ship pivots.

These are investigated by using the methods of Chapter VIII, illustrated in Fig. 34, except that instead of the wave profile the inclined waterline is used.

Problems

1. Plot the following data. Determine (a) the travel at which pivoting occurs, and (b) the poppet pressure.

Launching weight			3810 tons	
Moment of weight about fore poppet			567,000 foot-tons	
Travel, feet	295	359	391	423
Displacement, tons	1120	2430	3240	4090
Moment of displacement about fore poppet	232,800	453,400	577,900	707,000

2. At a given moment in a launching assume:

Moment against tipping	20,000 foot-tons
Launching weight	3810 tons
Displacement	2700 tons
Length of ways in contact	64 feet
Width of each sliding way	2.75 feet

What is the corresponding way-end pressure?

Part II

DESIGN

Chapter XIII

OWNER'S REQUIREMENTS, PRELIMINARY DIMENSIONS, POWER AND WEIGHT

REFERENCES

(*a*) Lovett, "Applied Naval Architecture," Chapter III.
(*b*) Taylor, "Speed and Power of Ships."
(*c*) "Principles of Naval Architecture," Volume I, Chapter III.

OWNER'S REQUIREMENTS

Assume that the prospective owners require a ship of the following characteristics:

Cargo deadweight	8000 tons
Passengers	50
Speed at sea	15 knots
Cargo cubic	60 cubic feet per ton of cargo
Maximum draft	27 feet
Propelling machinery	Geared turbines, watertube boilers, amidships
Oil fuel	for 10,000-mile radius

Extent of erections not specified. Crew, about 70

The determination of the above characteristics is an involved study in itself. It is a problem in operating costs, and the most profitable size, type and speed of ship is worked out by the owners either by

(a) analysis of the accounts of the various ships under their operation, or

(b) application of their operating cost data to an assumed series of sizes and speeds, resulting in curves or figures showing the most profitable size or speed. For

examples of such a study, see the paper in the *Transactions* of The Society of Naval Architects and Marine Engineers for 1923, called "Factors Affecting the Economy of Lake Freighters," or the chapter in Robert W. Morrell's "Oil Tankers" headed "Economical Speed for Tankers." Although the naval architect sometimes must make studies of this kind for the owners, they must in each case be based on their particular trade using operating cost data that directly applies to that trade. We will simply accept the above figures as the "owner's requirements."

In the following work, numerous places will be left blank, to be filled in by the student. Naturally, different men may arrive at slightly different values, and at the end of the chapter suitable values for the various items will be given, in order to permit going ahead on a common basis. The student can, if he chooses, simply turn to the end of the chapter and obtain the missing figures; but, if he first obtains his own figures, he will be sure that he is not skipping something.

Total Deadweight

The deadweight-displacement ratios given in the various references of Chapter VI and in Fig. 30 refer to the total deadweight, not the cargo deadweight, whereas our owner has specified cargo deadweight. So we must first approximate the other deadweight items: fuel, water, stores, complement, etc. This will be a guess, but a guess must be made in order to get started. If we make a bad guess, we will find it out before we finish this chapter and correct it. We will assume that the deadweight items other than cargo will be 2000 tons, bringing the total deadweight to 10,000 tons, tentatively.

Displacement

The approximate ratio of deadweight to displacement will be, from Fig. 30, for primarily freight vessels, about 0.70. Reference (*c*), page 120, gives for cargo vessels a deadweight coefficient of from 0.65 to 0.75.

This coefficient on the example ship was [ ]. The proposed ship, being somewhat faster, should have a slightly lower coefficient.

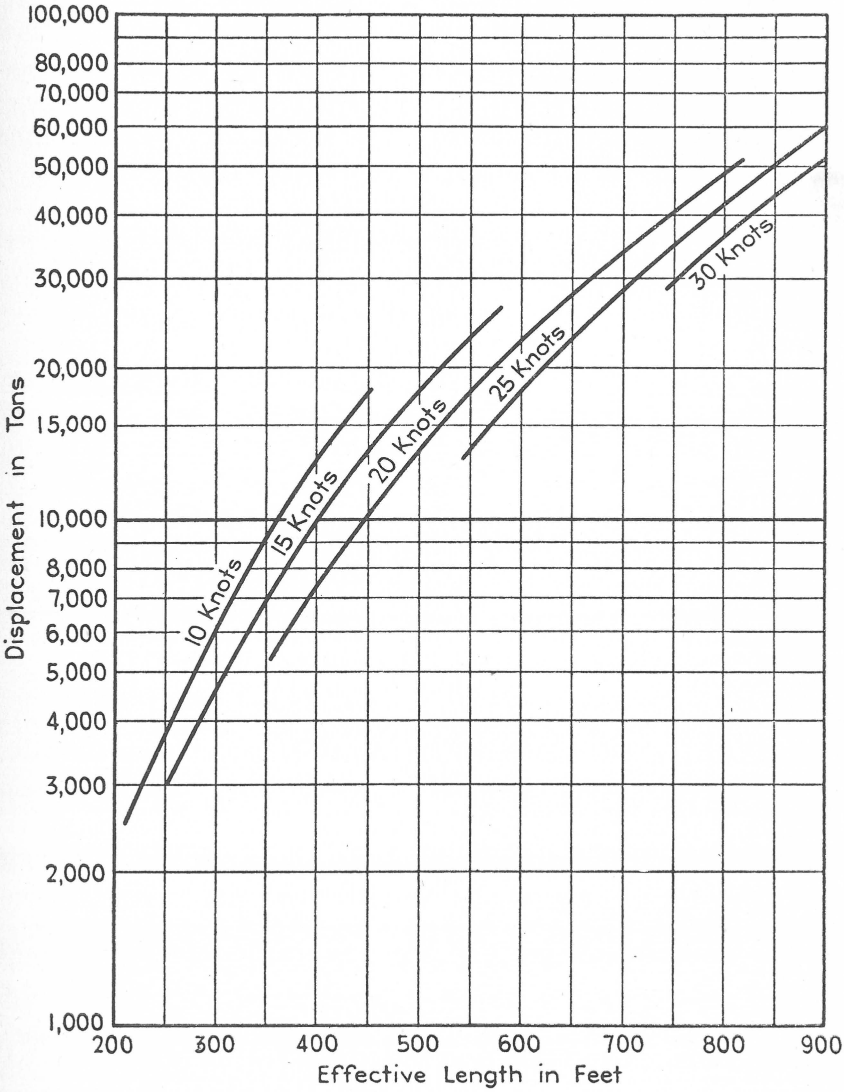


Fig. 56.—Relation between displacement, speed and length; normal merchant practice

It appears that our deadweight-displacement coefficient will be about [ ], giving a displacement of [ ] tons. After filling in above spaces, refer to the end of the chapter and substitute the values there given.

Tentative Length

Tentative length is chosen to give suitable values to two coefficients: speed-length ratio and displacement-length coefficient. This requires a "suitable" relationship between displacement, length and speed. Fig. 56 approximately represents good practice throughout the range covered. For 15,000 tons and 15 knots this chart indicates a trial length of about [ ] feet, but since length is the most expensive dimension of a ship, we will choose a slightly shorter length, namely [ ], which will give us a speed-length ratio of [ ] and a displacement-length ratio of [ ].

Fineness

For a given value of speed-length ratio the suitable value of the prismatic coefficient is closely limited. If this "suitable" value is exceeded, the resistance quickly becomes excessive. If the prismatic is made smaller, the ship becomes unnecessarily large. For ordinary merchant vessels, the expression: Prismatic coefficient $= (1.15 - 0.6V/\sqrt{L})$ represents good practice and for a given prismatic coefficient gives approximately the speed at which the resistance begins to increase rapidly.

The low prismatic associated with high speed results in narrow end holds which are poorly suited to stow cargo, and a fast freighter is obliged to strike a compromise between these two considerations. On the other hand, a fast passenger liner carrying but little cargo can use as fine a prismatic coefficient as the speed calls for.

The above expression gives a prismatic coefficient of about [ ] as good practice for our speed-length ratio. This determines our midship area, for

$$\text{Midship area} = \frac{35 \times \text{displacement}}{L \times \text{prismatic coefficient}}$$

$$= \frac{35 \times 15{,}000}{[\quad] \times [\quad]} = [\quad] \text{ sq ft}$$

Beam and Draft

For a given midship area the product of beam and draft determines the midship section coefficient. There is a wide difference in practice as to this coefficient. Values for cargo ships will be found varying from about 0.99 down to perhaps 0.90 (as for the arc-form type of lines brought out by Sir Joseph Isherwood). The effect on resistance of wide variations of midship section coefficient seems to be small [see Reference (*b*)]. In general, the draft should be the deepest allowable and, pending an investigation of stability, a beam equal to $L/10$ plus 15 is good practice. This gives us a beam of 60 feet, a draft of 27 feet, and a midship rectangle area of 1620 square feet, which is too close to the midship section area required above. We cannot increase the draft, so we will increase the beam to 61 feet, which gives us a midship rectangle area of 1647 square feet, a midship coefficient of [ ] and a block coefficient of [ ].

Depth

Depth will be determined later by the freeboard rules and by an estimate of the volume of the cargo spaces. A tentative depth will be chosen now by the ratio of length to depth given by the American Bureau of Shipping rules. These rules are for a depth of between $L/14$ and $L/14$ plus 5; that is, from 32 to 37 feet. Until we have later checked the freeboard and cargo capacity, we will assume a depth of 33 feet.

Power

For preliminary power we may refer to Fig. 57 which gives, for 15,000 tons displacement and 15 knots, about [ ] effective horsepower. Or we may make an inde-

pendent estimate, as follows, using Taylor's standard series [Reference (*b*)]:

Frictional resistance, in pounds per ton displacement, from Fig. 69, of Reference (*b*), neglecting corrections..........	[ ]
Residual resistance, in pounds per ton displacement, from Figs. 74 and 75, of Reference (*b*), interpolating.........	[ ]
Total resistance per ton............................	[ ]
$ehp = \frac{R}{\Delta} \times \Delta \times V \div 326 =$ ([ ] × 15000) × 15÷ 326 =..	[ ]

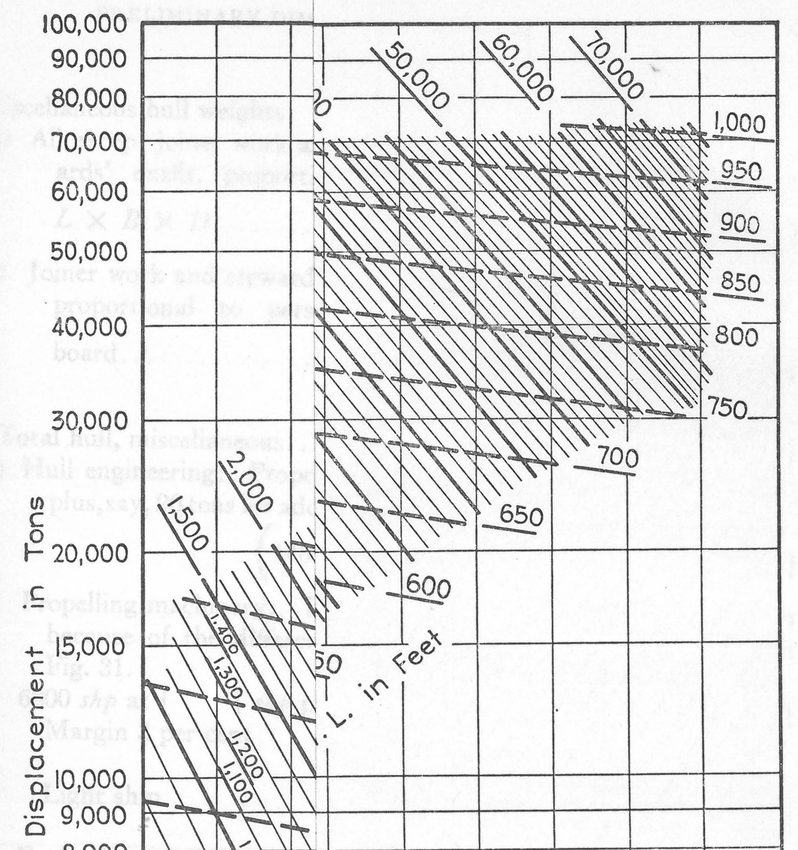
Evidently, bare hull *eph* will be about.............	[ ]
eph, including appendages, 1.03 × [ ].........	[ ]
Using a propulsive coefficient of 0.60 gives........	[ ] *shp*
Using a margin of 15 per cent gives, for sea power..	[ ] *shp*

At this point refer to the end of the chapter and change the various dimensions, etc., which have been written in the blank spaces, to those given there.

We may now make our second estimate of the weight of the loaded ship, as described on page 79, to see if our tentative displacement of 15,000 tons is good. In place of actual weights from ship data, we will use the weights of the example ship as tabulated in Chapter I, and proportion our weights and vertical centers of gravity from them. Letters "(a)," etc., refer to the weight groups given on page 79.

Cubic number coefficient, for example ship (Chapter VI, Problem 2)..	[ ]
Corrected for difference in block (Chapter VI)............	[ ]
Corrected for difference in L/D (negligible)	

	Tons	*VCG*
Main hull steel: $\frac{450' \times 61' \times 33'}{100} \times$ [ ] =	[ ]	[ ]
Erections: Proportional to $L \times B$, plus, say, 30 tons for added passengers.....	[ ]	[ ]
Houses and masts: Add, say, 40 tons for added passengers.............	[ ]	[ ]
(a) Total steel................................	[ ]	[ ]



Fir with "suitable" *LWL*

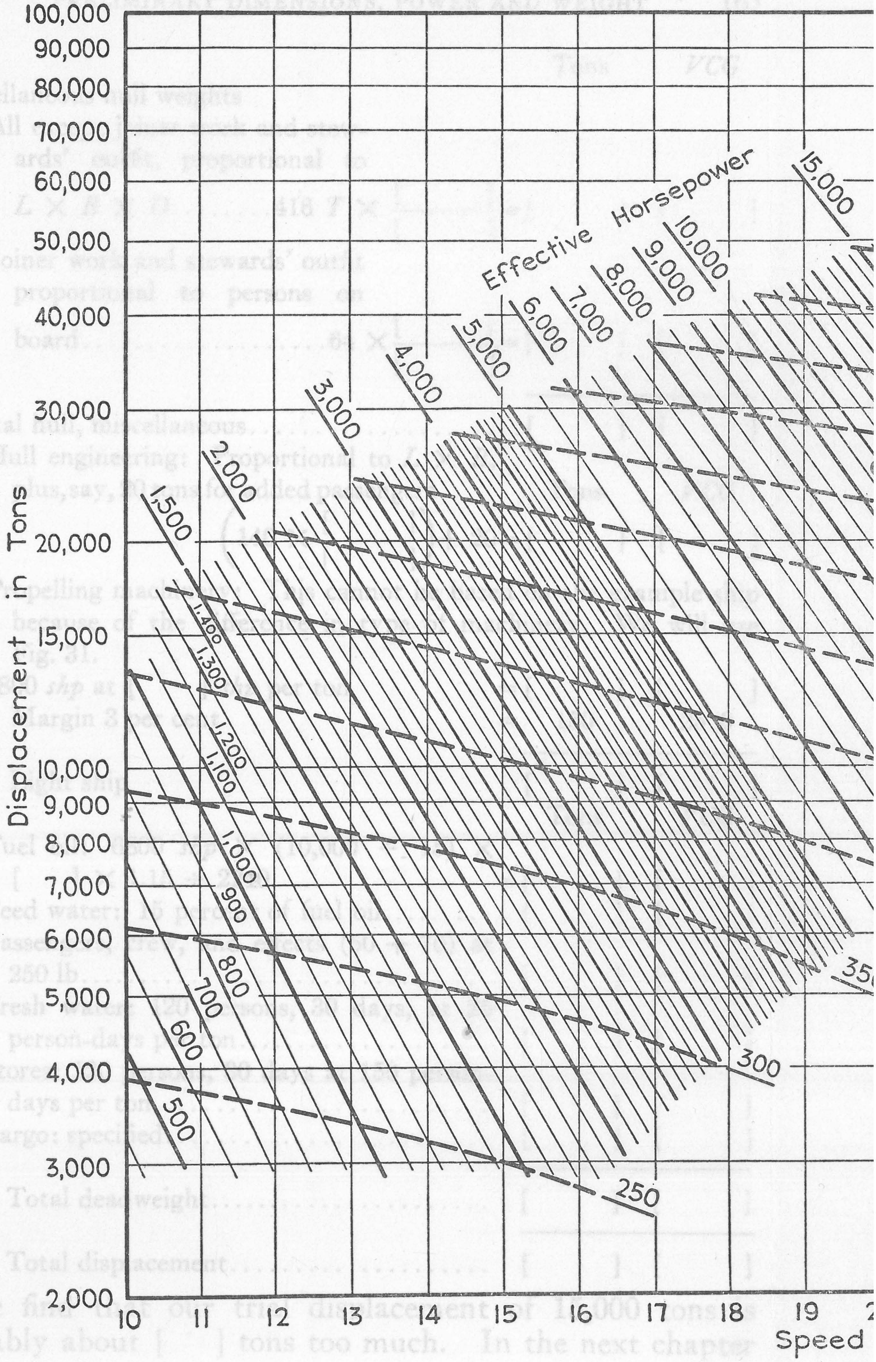
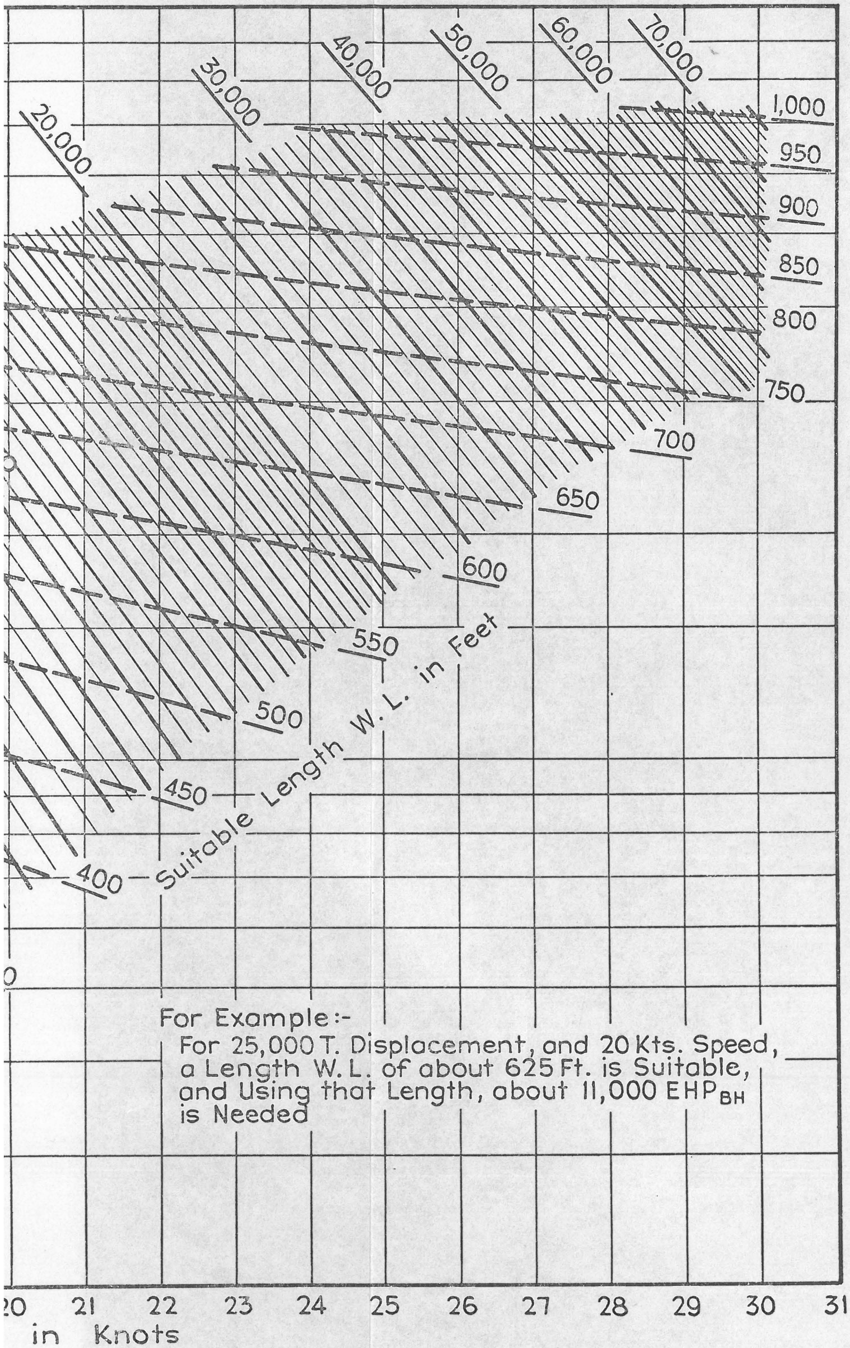


Fig. 57.—Approximate effective horsepower curves, for norma



l merchant vessels, (bare hull) together with "suitable" *LWL*

Item	Tons	*VCG*
Miscellaneous hull weights		
(b) All except joiner work and stewards' outfit, proportional to $L \times B \times D$........416 $T \times \frac{[\quad\quad]}{[\quad\quad]} =$	[ ]	[ ]
(c) Joiner work and stewards' outfit proportional to persons on board..................64 $\times \frac{[\quad\quad]}{[\quad\quad]} =$	[ ]	[ ]
Total hull, miscellaneous....................	[ ]	[ ]
(d) Hull engineering: Proportional to $L \times B$, plus, say, 20 tons for added passengers	Tons	*VCG*
$\left(140 \times \frac{[\quad\quad]}{[\quad\quad]}\right) + 20 =$	[ ]	[ ]
(e) Propelling machinery: This cannot be based on the example ship because of the difference in type of machinery. We will use Fig. 31.		
6800 *shp* at [ ] *shp* per ton =	[ ]	[ ]
Margin 3 per cent =	150	25.0
Light ship	[ ]	[ ]
	Tons	*VCG*
(f) Fuel oil: 6800 *shp* $\times$ (10,000 $\div$ 15) $\times$ [ ] $\times$ 1.15 $\div$ 2240 =	[ ]	[ ]
Feed water: 15 percent of fuel oil.........	[ ]	[ ]
(g) Passengers, crew, and effects (50 + 70) at 250 lb...............................	[ ]	[ ]
(h) Fresh water: 120 persons, 30 days, at 25 person-days per ton....................	[ ]	[ ]
Stores: 120 persons, 30 days at 150 person-days per ton.........................	[ ]	[ ]
(i) Cargo: specified........................	[ ]	[ ]
Total deadweight.....................	[ ]	[ ]
Total displacement..................	[ ]	[ ]

We find that our trial displacement of 15,000 tons is probably about [ ] tons too much. In the next chapter

we will adjust the dimensions accordingly, check freeboard and stability, and carry the design into its next stage.

All the blank spaces in the preceding work should now be checked against the data given below:

Deadweight coefficient, example ship	0.69
Assumed deadweight coefficient, problem ship	0.67
Trial displacement, tons	15,000
Trial length, from Fig. 59, feet	465
Trial length, used, feet	450
Speed-length ratio	0.71
Displacement-length ratio	164
Prismatic coefficient, good practice	0.72
Midship area, square feet	1620
Midship coefficient	0.980
Block coefficient	0.707
Power, from Fig. 57	3350
Frictional resistance, from Taylor, pounds per ton	3.45
Residuary resistance, from Taylor, pounds per ton	1.57
Total resistance, from Taylor, pounds per ton	5.02
Power by Standard Series	3500
Probable *ehp*, bare hull	3400
Probable *ehp*, including appendages	3500
Probable *shp*	5900
Probable *shp*, sea power	6800
Cubic number coefficient, example ship	0.289
Cubic number coefficient, problem ship	0.283

Preliminary weights and centers, problem ship

	Tons	*VCG*
Main hull steel	2570	17.1
Erections	280	40.8
Houses and masts	200	46.0
Total steel	3050	21.2
Miscellaneous hull weights (b)	550	25.0
Miscellaneous hull weights (c)	170	39.0
Total hull miscellaneous	720	28.3
Hull engineering	190	30.0
Propelling machinery	730	16.1
Light Ship	4840	22.0

Light ship	4840	22.0
Fuel oil (0.6 lb per *shp*-hr)	1400	10
Feed water	210	3
Passengers, crew and effects	10	36
Fresh water	150	13
Stores	20	30
Cargo	8000	20
Total deadweight	9790	18.1
Total displacement	14,630	19.4
Excess trial displacement	370	

Chapter XIV

REVISED DIMENSIONS, FREEBOARD AND STABILITY, CUBIC CAPACITY, SKETCH PROFILE, FLOODABLE LENGTH

Reference

(*a*) Load Line Regulations of the United States (1941).

Chapter XIII ended with a displacement about 370 tons (2 percent) too large. We can reduce this by reducing either length, beam, draft or fullness. Before we decide which to do, we should check freeboard (which may change depth and, with it, the weight, displacement and *VCG*) and stability (which may change the beam).

Freeboard

Obtain the load line rules, Reference (*a*). Refer to page 127 and copy the arrangement of the solution to Problem 1. Assume that our bridge will be 165 feet long (to allow for passenger arrangements), forecastle 38 feet and poop 43 feet (proportioned from example ship), all with openings as in example ship. Assume also standard sheer and camber.

Table freeboard for 450-foot *LBP*.................... [ ] in.

Block coefficient correction $\dfrac{[\quad\quad] + 0.68}{1.36} \times [\quad\quad] = [\quad\quad]$

Depth correction $\left([\quad\quad] - \dfrac{450}{15}\right) \times 3''$ [ ]

[ ] in.

Effective length of erections [ ] ft., or [ ] percent *L*

Correction for erections [ ] percent of 42″ [ ]

Summer freeboard ([ ] ft) [ ] in.

Draft [ ] − [ ] [ ] ft.

Evidently to use a draft of 27 feet we must increase our depth. Trying a depth of 34 feet shows that it would just about permit a draft of 27 feet. (Check this.) We will accordingly change our depth to 34 feet.

We must next check stability, but before we can do this

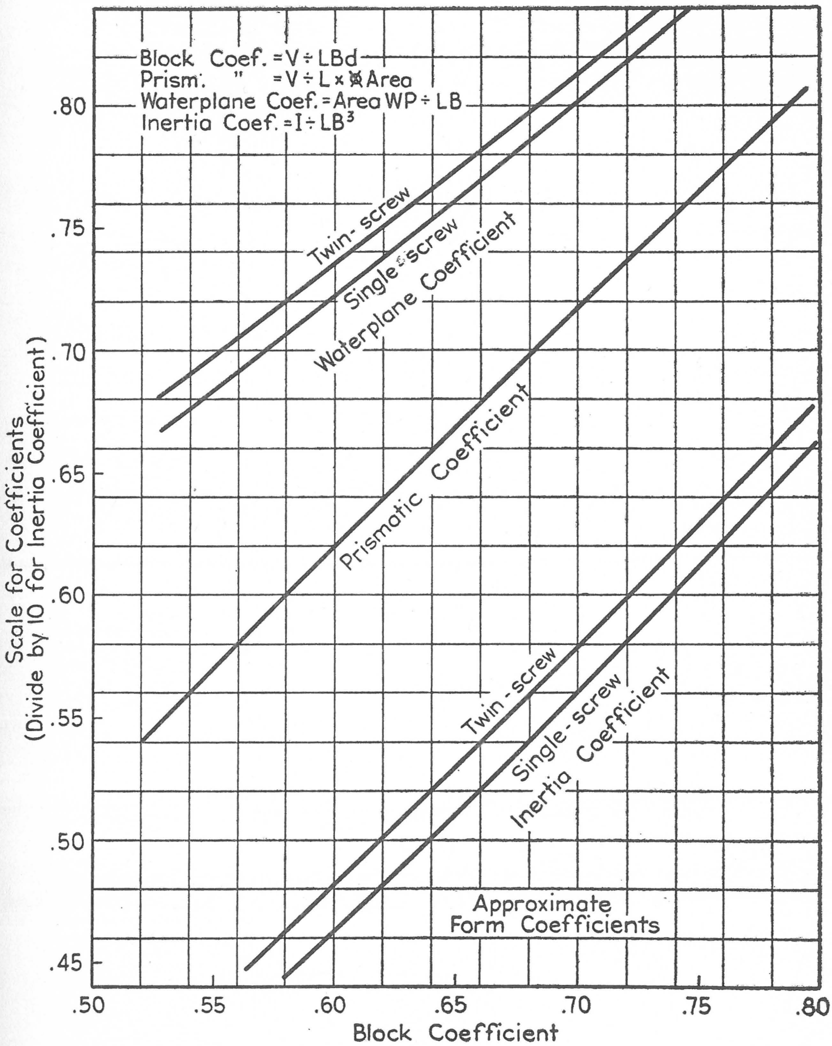


Fig. 58

we must correct the weight and *VCG* to the new depth of 34 feet. This increase in depth will increase our steel hull weight by about $\sqrt{34/33}$, or from 2570 to [ ] tons. The other steel weights will not change, but we must add about 10 tons for other weights which will be increased by increased depth, making our light weight [ ] tons heavier than on page 163, or [ ] tons. The deadweight will not be changed, so the total loaded weight will be [ ] tons; still [ ] tons less than the tentative displacement.

The *VCG* may be assumed to be raised in proportion to the depth, to 19.4 feet × 34/33, or to 19.9 feet.

Check on Stability in Load Condition (See Chapter III)

To find *GM* we must approximate *KM*, which is (see Fig. 8) *VCB* + *BM*. We have a preliminary block coefficient of 0.707. If we can assume approximate values of waterplane coefficient and inertia coefficient, we can make a close approximation to *KM*. The inertia coefficient has not been mentioned before, but is $I \div LB^3$. For example, for a rectangular waterplane it would be $^1/_{12}$, or 0.0833. Fig. 58 shows approximately how the various form coefficients vary with the block coefficient, and gives us a guide as to our probable coefficients, even though we have not yet reached the stage of drawing lines. From this figure we see that a block of 0.707 will be associated with a waterplane coefficient of about [ ] and an inertia coefficient of about [ ].

The *VCB* will be very nearly

$$\text{Draft} \times \frac{\text{waterplane coefficient}}{\text{waterplane coefficient} + \text{block coefficient}}$$

(This simple formula for approximate *VCB* is more accurate than several other more complicated ones.) This gives our *VCB* equal to

$$27 \times \frac{[\quad\quad]}{[\quad\quad] + 0.707} = [\quad\quad] \text{ feet}$$

The *BM* will be

$$\frac{B^2}{\text{draft}} \times \frac{\text{inertia coefficient}}{\text{block coefficient}}$$

(This can be derived from the formula $BM = I/V$ given on page 22, and the ratio of inertia coefficient to block coefficient is the "factor" referred to on page 37.)

Our approximate BM is therefore

$$\frac{(61)^2}{27} \times \frac{[\quad]}{0.707}, \text{ or } [\qquad] \text{ feet}$$

and

$$KM = VCB + BM \text{ or } [\qquad] \text{ feet}$$

We should allow a small margin in VCG, say, 0.5 foot. For an elaborate passenger liner a larger allowance would be made. We should also make an allowance for free surface (see Chapter III). As we have as yet no arrangement plans, we cannot calculate this, but 0.5 foot is reasonable. This gives a VCG of [ ] including free surface and margin, and a GM of [ ] feet.

Refer now to the end of this chapter and check the missing figures.

This GM is too high. Our vessel is a passenger ship and should not have a GM much over $0.05B$, or about 3 feet (see Chapter III, page 45).

Reducing beam, without changing form coefficients, does not change VCB but reduces BM in proportion to $(B)^2$. To reduce GM one foot, BM must be reduced from 11.1 feet to 10.1 feet, which would require reducing the beam from 61 feet to about 58 feet. This would reduce the displacement too much, and since the draft cannot be increased, the length would have to be increased. This would in turn increase freeboard, depth and VCG and further reduce GM. We will try a smaller reduction in beam, and an increase in length and depth, namely,

460 feet × 59 feet × 34 feet 3 inches

New freeboard

Table freeboard for 460 feet........................	[ ]	in.
Fullness correction [ ] $\times$ (0.707 + 0.68)/1.36...	[ ]	
Depth (34.30 − 460/15) $\times$ 3.....................	[ ]	
Superstructure correction is now....................	[ ]	
New freeboard [ ] feet......................	[ ]	in.

Draft 34.30′ − [ ] ≐ [ ] feet

This is close enough.

Weight

New cubic number 460 $\times$ 59 $\times$ 34.25 = 9300
Old cubic number 450 $\times$ 61 $\times$ 34 = 9330

This difference is too small to change the weight for, but we will add 20 tons for increased length-depth ratio and the total *VCG* can be assumed to rise in proportion to the depth, giving a light ship of 4910 tons and a loaded weight of [ ] tons, *VCG* [ ] feet.
The new block coefficient will be

$$([\quad] \times 35) \div (460 \times 59 \times 27) = [\quad]$$

and the prismatic can still be 0.72

VCB will still be about..........................	[ ]	ft.
BM will now be $59^2/27 \times 0.0563/0.702$ =	[ ]	ft.
KM will be....................................	[ ]	ft.
VCG including margin and free surface.............	21.1	ft.
GM..	[ ]	ft.

This is still a little high, but will be accepted. The period of roll (see Fig. 16) will be about [ ] seconds.

Cargo Cubic

Our owners have specified 60 cubic feet of cargo space per ton of cargo; that is, about [ ] cubic feet. Before going further we must see if our tentative dimensions will probably comply with this, basing our approximation on

the example ship. Our ship differs from the example ship in that it is larger, finer and higher powered.

If we proportion our cargo space directly to $L \times B \times D \times$ waterplane coefficient, we would get

$$400{,}000 \times \frac{460 \times 59 \times 34.5 \times 0.802}{420 \times 54 \times 30.25 \times 0.862} = [\qquad\qquad] \text{ cubic feet}$$

This automatically includes an increase of machinery and bunker space proportional to the size of the ship, that is, in the ratio of 504,000 ÷ 400,000, or 1.26, and allows 24,000 cubic feet for increased machinery and bunkers, which will probably be sufficient. To satisfy ourselves further, we find from similar ships of similar speed that cargo cubic is about 0.53 *LBD*, which would be for us [ ] cubic feet. Evidently we will have sufficient cargo space, and are right in adopting a "three island" ship (one with poop, bridge and forecastle above the freeboard deck) rather than a shelter-deck type. If the owner had specified a high stowage rate for a light cargo, such as 100 cubic feet per ton, or 800,000 cubic feet, we should have had to go to a "complete superstructure" type of ship, and perhaps increase the depth to more than that required by freeboard, to get the required capacity. Then we should have had to start all over again.

Trial Lengths

At this stage designers sometimes choose a series of lengths both less and more than the tentative length, as, for instance, 440 feet and 480 feet for our ship, and investigate the effect on steel weight, resistance, cargo cubic and other factors. The steel weight is varied by modifying the cubic number coefficient for block coefficient and length-depth ratio (page 80). The change in prismatic coefficient and wetted surface will change the power. Added length will increase the frictional resistance and will usually reduce the residual resistance.

It is easier to make such an investigation than it is to decide on the best length by studying the results, for they are frequently contradictory; for instance, the shorter ship may require less steel but more power. We shall not include a series of trial lengths in this design, but shall accept our tentative length of 460 feet. It should, however, be noted that although we tried to use a shorter length than that indicated by "good practice" as shown in Fig. 56, we were forced back toward that length to obtain satisfactory characteristics.

Sketch Profile, and Floodable Length

We must now draw a sketch of the arrangement of the ship showing holds, decks, bulkheads, erections and general assignment of spaces. This will permit us to make the more accurate estimate of weights described in the next chapter, and permit a preliminary check on the floodable length. Such a profile is shown on Fig. 59.

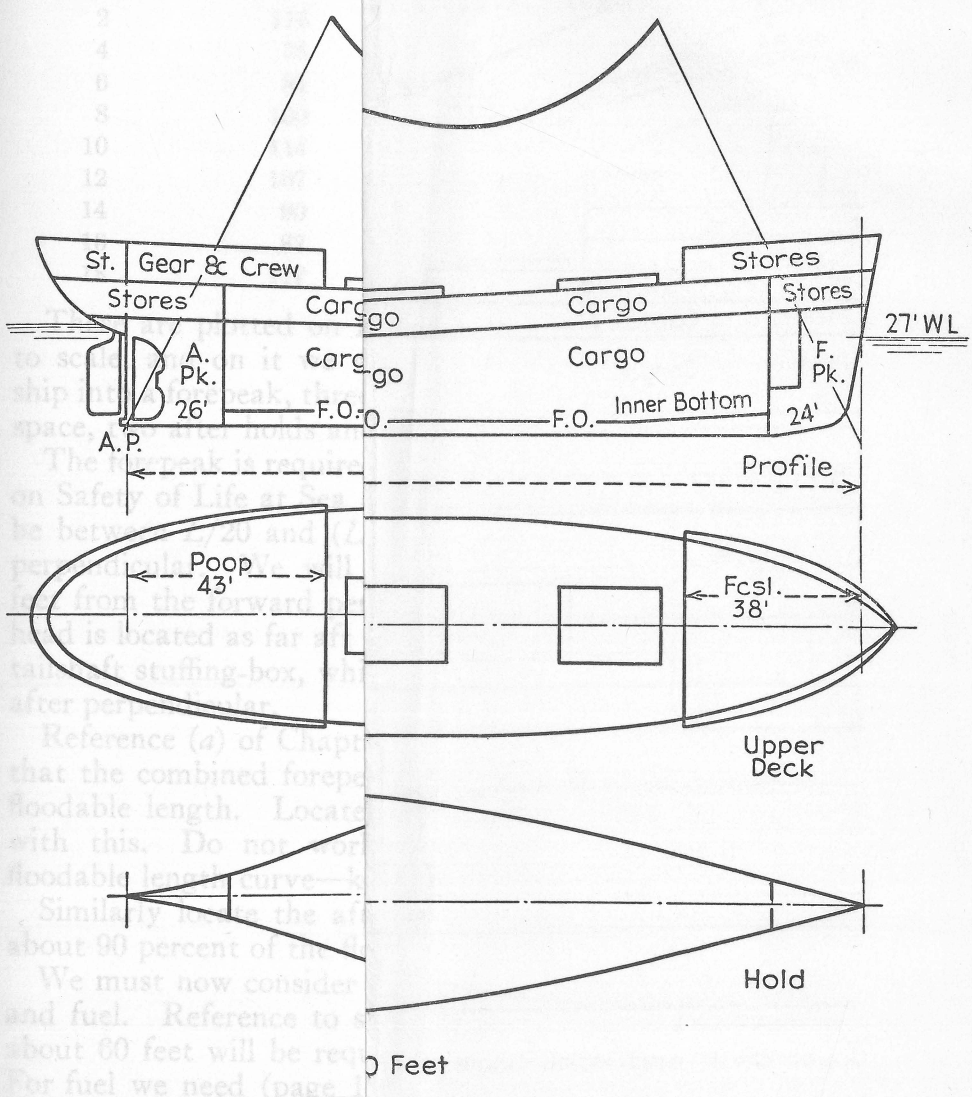
A preliminary floodable length curve is now needed to locate the bulkheads. An exact curve cannot be calculated at this stage, but an approximate curve can be obtained, if a curve for a similar ship is available, by multiplying the floodable length of the similar ship by the ratio:

$$\frac{\text{new } LBP \times \text{new freeboard/depth} \times \text{old permeability}}{\text{old } LBP \times \text{old freeboard/depth} \times \text{new permeability}}$$

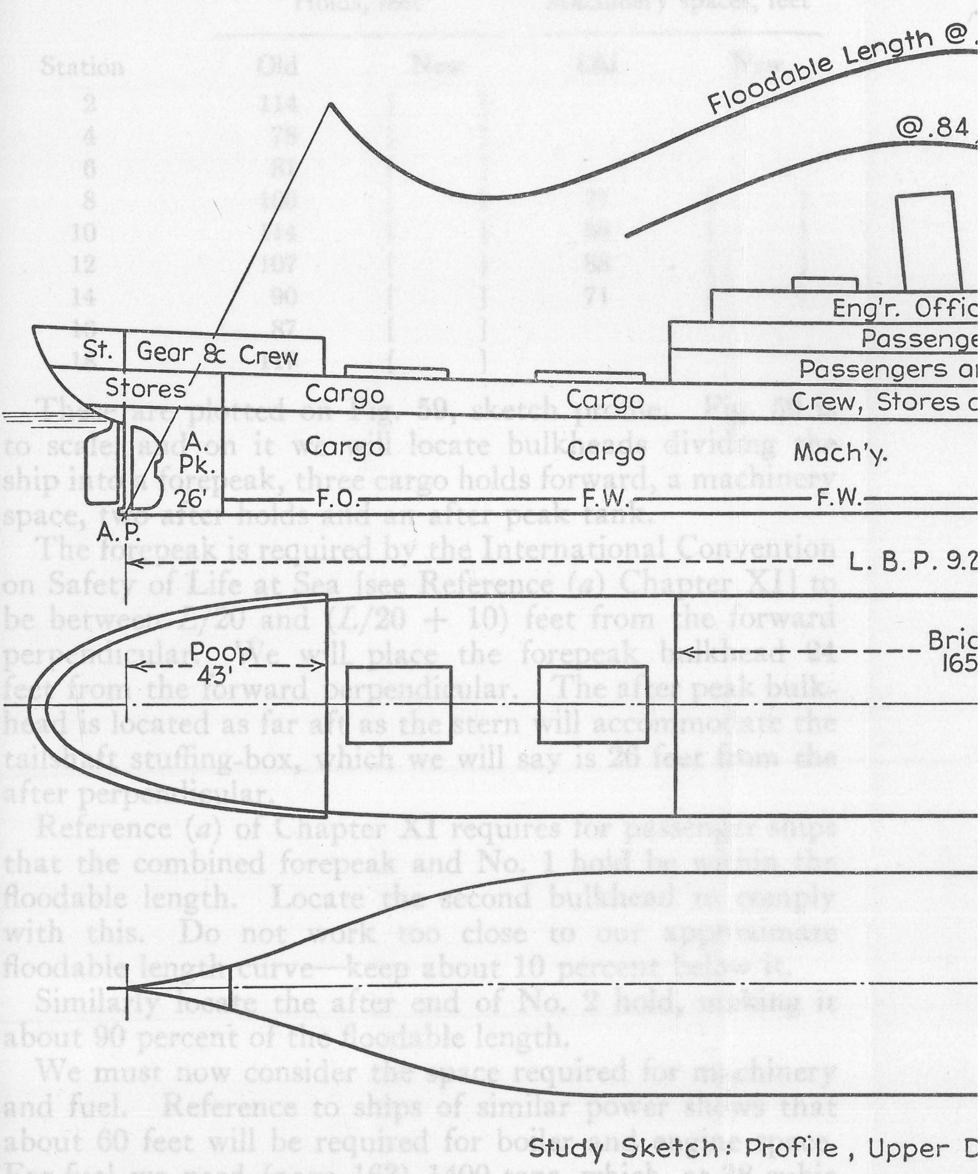
We will proportion such a curve from that given for the example ship in Fig. 41. Since there may be some living spaces below the bulkhead deck in our ship, because of passengers, the permeability of the new ship is assumed to be 5 percent more than for the example ship, and the conversion factor for floodable lengths will be

$$\frac{460 \times (7.25/34.25) \times 1.00}{420 \times (6.25/30.25) \times 1.05} = [\quad]$$

Floodable lengths for the example ship, and the corresponding lengths for the new ship are as follows:

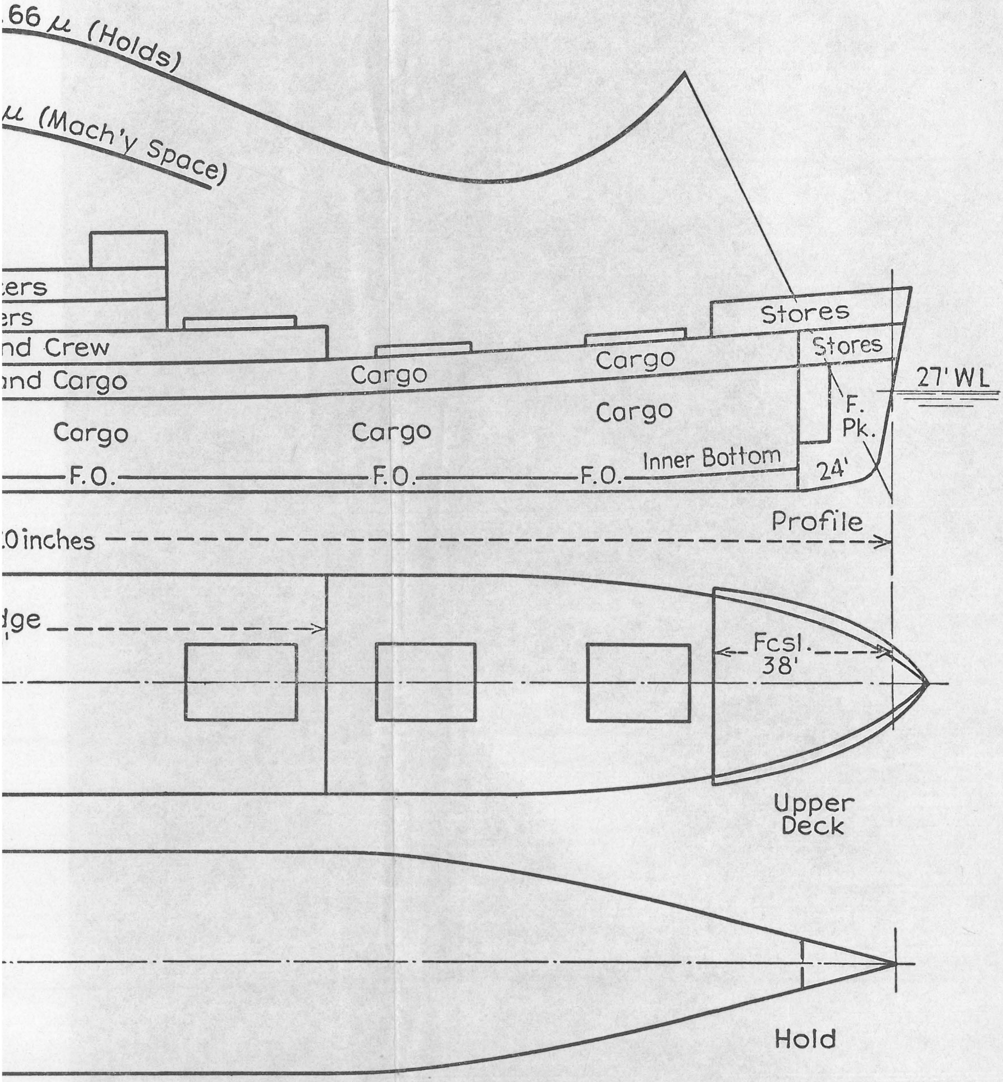


St.
Gear & Crew
Stores
Cargo
Stores
Stores
27' WL
A.
Pk.
26'
Cargo
Cargo
F.
Pk.
F.O.
Inner Bottom
F.O.
24'
A.P.
Profile
Poop
43'
Fcsl.
38'
Upper
Deck
Hold



Floodable Length @
@.84
Eng'r. Offic
Passenge
Passengers a
Crew, Stores a
Mach'y.
St.
Gear & Crew
Stores
Cargo
Cargo
Cargo
Cargo
A.
Pk.
26'
A. P.
F.O.
F.W.
F.W.
L. B. P. 9.2
Poop
43'
Bric
165
Study Sketch: Profile, Upper D

Fi



Deck and Hold. Scale: 1" = 50 Feet

g. 59

	Holds, feet		Machinery spaces, feet	
Station	Old	New	Old	New
2	114	[ ]		
4	78	[ ]		
6	81	[ ]		
8	100	[ ]	77	[ ]
10	114	[ ]	90	[ ]
12	107	[ ]	88	[ ]
14	90	[ ]	71	[ ]
16	87	[ ]		
18	117	[ ]		

These are plotted on Fig. 59, sketch profile. Fig. 59 is to scale, and on it we will locate bulkheads dividing the ship into a forepeak, three cargo holds forward, a machinery space, two after holds and an after peak tank.

The forepeak is required by the International Convention on Safety of Life at Sea [see Reference (*a*) Chapter XI] to be between $L/20$ and $(L/20 + 10)$ feet from the forward perpendicular. We will place the forepeak bulkhead 24 feet from the forward perpendicular. The after peak bulkhead is located as far aft as the stern will accommodate the tailshaft stuffing-box, which we will say is 26 feet from the after perpendicular.

Reference (*a*) of Chapter XI requires for passenger ships that the combined forepeak and No. 1 hold be within the floodable length. Locate the second bulkhead to comply with this. Do not work too close to our approximate floodable length curve—keep about 10 percent below it.

Similarly locate the after end of No. 2 hold, making it about 90 percent of the floodable length.

We must now consider the space required for machinery and fuel. Reference to ships of similar power shows that about 60 feet will be required for boiler and engine space. For fuel we need (page 163) 1400 tons, which, at 38 cubic feet per ton with 5 percent for expansion, is 56,000 cubic feet. Assume that, as shown in Fig. 59, four of the six double bottoms can be used for fuel oil, with a combined

capacity of about 350 feet × 3.5 feet × say 40 feet or 49,000 cubic feet. The rest can be carried in wing tanks and settling tanks in the machinery space, but to avoid crowding this space unduly we will increase its length to 65 feet.

The aftermost hold will be made of such a length that it, together with the after peak tank, will be within the floodable length. This is not required by law, but it is desirable and in this case there is no reason why it should not be done.

Now go back and adjust the bulkheads so that all compartments (except the machinery space) are about the same percentage of the floodable length. This gives the highest degree of safety possible with the number of bulkheads used. The resulting compartment length should now be about as tabulated below:

	Feet
Forepeak	24
No. 1 hold	68
No. 2 hold	75
No. 3 hold	60
Machinery space	65
No. 5 hold	72
No. 6 hold	70
After peak	26
Total	460

Draw lines representing these bulkheads on Fig. 59, located to scale, extending from base line to upper deck on the profile, and from side to side on the hold plan.

This sketch plan will be subject to revision when the engineers design the machinery and when the arrangement plans are developed. In the meantime it will serve as a basis of the weight estimate which follows in the next chapter. The deck plans are drawn by eye, and are to be revised to suit the actual lines developed in the next section. The extent of houses is made roughly similar to other ships carrying about 50 passengers, and will be changed as necessary when arrangements are drawn.

Data which have been omitted in this chapter, for the student to fill in, are as follows:

Table freeboard	87.1 inches
Corrected for fullness	88.8 inches
Depth correction	+9.1 inches
Effective length of erection	246 feet or 55 percent
Correction for erections (41 percent)	17.2 inches
Summer freeboard	80.7 inches
Allowable draft	26.33 feet
Revised steel weight	2610 tons
Increase in light weight	50 tons
New loaded weight	14,680 tons
Difference between weight and Δ	320 tons
Waterplane coefficient	0.807
Inertia coefficient	0.0568
VCB	14.4 feet
BM	11.1 feet
KM	25.5 feet
VCG	20.9 feet
GM	4.6 feet
New table freeboard	90.2 inches
Corrected for fullness	92.0 inches
Depth correction	+10.9 inches
Correction for erections (39.5 percent)	−16.6 inches
New freeboard	7.19 feet
New draft	27.11 feet
New loaded weight	14,700 tons
New *VCG*	20.1 feet
New block coefficient	0.702
New *VCB*	14.4 feet
New *BM*	10.3 feet
New *KM*	24.7 feet
New *GM*	+3.6 feet
Period of roll	13 seconds
Required cargo cubic	480,000 cubic feet
Proportioned cargo cubic	504,000 cubic feet 490,000 cubic feet
Floodable length ratio	1.07

New floodable lengths

Station	Holds, feet	Machinery spaces, feet
2	122	
4	83	
6	87	
8	107	82
10	122	96
12	114	94
14	96	76
16	93	
18	125	

Chapter XV

REVISED WEIGHT AND DISPLACEMENT, FINAL DIMENSIONS, POWER, LINES

We will now make our third weight estimate as described in Chapter VI, page 82, using the revised dimensions and the sketch profile and deck plan obtained in the preceding chapter. The various unit weights in the following are assumed and are reasonable for this ship, but in practice would be taken from the most similar ship available for reference. To go actually through all the weight groups would take an unnecessary length of time, so some typical groups will be taken up, with the understanding that the others would be similarly treated.

The principal dimensions of the ship are now:

Length between perpendiculars	460 feet 0 inches
Beam	59 feet 0 inches
Depth	34 feet 3 inches
Draft	27 feet 0 inches
Block coefficient	0.702
Waterplane coefficient	0.80

STEEL WEIGHT

Main hull

	Tons
Shell and framing, $(B + 2D) \times L \times 0.88$ at 54 lb per sq ft $\div$ 2240	[ ]
Decks, including stanchions and girders, $2 \times L \times B \times 0.83$ at 34 lb per sq ft	[ ]
Inner bottom, $L \times B \times 0.50$ at 29 lb per sq ft	[ ]
Main bulkheads, $7 \times B \times D \times 0.75$ at 28 lb per sq ft	[ ]
Stem and stern, from reference ships	16
Miscellaneous bulkheads and casings, from reference ships	100
Foundations, from reference ships	50
Miscellaneous steel items, from reference ships	150
Rivet heads and tolerance 4 percent	100
Total main hull (cubic number coefficient [ ])	[ ]

Erections

Forecastle, 38′ at 0.90 ton per linear foot	[ ]
Bridge, 165′ at 1.10 tons per linear foot	[ ]
Poop, 43′ at 1.05 tons per linear foot	[ ]
Total erections	[ ]
Houses, about 100,000 cu ft at 5.0 lb per cu ft	[ ]
Masts and spars, from reference ship	28
Total houses, etc.	[ ]
Total steel (cubic number coefficient [ ])	[ ]

CARPENTER WORK

Inner bottom ceiling, 335′ × 30′ at 11 lb	[ ]
Hold sparring, on bulkheads, etc. { 335′ × 32′ × 2 at 70 lb	[ ]
Hold sparring, on bulkheads, etc. { 4 × 40′ × 32′ at 7 lb	[ ]
Hatch covers, 30′ × 20′ × 10 at 9 lb	[ ]
Carpenter decks (sides only) { bridge deck, 2000 sq ft at 12 lb	[ ]
Carpenter decks (sides only) { boat deck, 2000 sq ft at 12 lb	[ ]
Miscellaneous carpenter items, from reference ship	25
Total carpenter work	[ ]

JOINER WORK

Decks: house top, 8000 sq ft at 5 lb	[ ]
Accommodations, 70 passengers and officers at 1500 lb each	[ ]
Cold stores insulation, from reference ship	16
Miscellaneous, from reference ship	20
Total joiner work	[ ]

STEWARD'S OUTFIT

Galley and pantry outfit, 120 persons at 90 lb	[ ]
Bedding and linen, 120 persons at 75 lb	[ ]
Furniture and stateroom outfit, 120 persons at 220 lb	[ ]
Miscellaneous	10
Total steward's outfit	[ ]

(Note that our combined figure for joiner work and steward's outfit (c), on page 163, missed this rather badly.)

Anchors, Chains and Lines

(American Bureau of Shipping rules are needed for this.)

Equipment Tonnage

(Section 29 of American Bureau of Shipping rules.)

Under freeboard deck, 460′ × 59′ × 34.25′ × 0.708 ÷ 100	[ ]
Poop, bridge and forecastle, 246′ × 59′ × 8′ × 0.75 ÷ 150	[ ]
First tier of houses, 120′ × 43′ × 8′ × 0.75 ÷ 200.......	[ ]
Second tier of houses, 120′ × 40′ × 7.5′ × 0.75 ÷ 250...	[ ]
Equipment tonnage..................................	[ ]

Equipment

Equipment Numeral (American Bureau of Shipping rules, Table 16)..	[ ]
Bower anchors, stockless, 2 at [ ] lb................	[ ]
Spare bower, stockless, 1 at [ ] lb..................	[ ]
Stream anchor, stockless, 1 at [ ] lb................	[ ]
Chain cable, [ ] fathoms [ ] in. at [ ] lb (Table 19)..	[ ]
Stream line wire, 90 fathoms, $1\frac{1}{2}''$ at 21.3 lb.............	0.9
Towlines, steel, 130 fathoms, $1\frac{3}{4}''$ at 29.1 lb..............	1.7
Hawsers, Manila, 2 × 90 fathoms, $8\frac{1}{2}''$ at 12.9 lb.........	1.0
Warps, Manila, 2 × 90 fathoms, $7\frac{1}{2}''$ at 10.8 lb..........	0.9
Miscellaneous rope outfit, from reference ships............	2.5
Total anchors, chains and lines........................	[ ]

We will assume that all the weight groups have been similarly estimated as well as is possible at this stage of the design and that the total light weight comes out to be 4950 tons instead of 4910 as given on page 170. This, with the previous total deadweight of 9790, will give us a displacement of 14,740 tons, whereas our tentative dimensions and coefficients give us a displacement of 14,700 tons (page 170). This difference is almost negligible and is taken care of by increasing the block coefficient to 0.704; the prismatic can stay at 0.72, giving a midship coefficient of 0.978.

We should now calculate a complete shaft horsepower

curve throughout the range of speeds we are interested in, say, from 13 to 17 knots, using Taylor's standard series.

In using the standard series we should not base our coefficients on the length between perpendiculars, but upon the "effective" length. This is a somewhat arbitrary average length of the underwater body, and, for the usual single-screw aperture type of stern and moderately raked bow, will be about 99 percent of the length between perpendiculars. We shall call our effective length 455 feet. This makes our effective prismatic about 0.728 and our displacement-length coefficient 156. Wetted surface, from Fig. 40, is 455 feet × 113 feet × 0.928 × 0.855, or [   ] square feet.

Using these values, and Taylor's standard series, fill out Table 4, "calculation for *shp*," and plot *ehp* and *shp* on Fig. 60. Table 4(a) at the back of the book shows the calculation. Note that at 15 knots the power, including

TABLE 4.—HORSEPOWER CALCULATION FOR PROBLEM SHIP

Displacement = 14,740 tons WS = 40,800 sq ft B/Dr = 2.19
Prismatic coefficient, 0.728 Displacement-length ratio, 156
Length correction to R_f (Fig. 39) for 455 feet = 0.995

		Rr, lb per ton							
$V/\sqrt{L}$	V, kts	2.25	3.75	2.19	ehp_r	$\frac{ehp_f}{1000 \text{ sq. ft.}}$	ehp_f	ehp_{BH}	shp_{sea}
0.60	12.87	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]
0.65	13.94	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]
0.70	15.01	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]
0.75	16.09	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]
0.80	17.16	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]	[ ]

$$ehp_r = Rr/\text{ton} \times \Delta \times V \div 325.7 = Rr/\text{ton} \times V \times 45.26$$

$$ehp_f = \frac{ehp_f}{1000 \text{ sq ft}} \times \text{length correction} \times \frac{WS}{1000} = \frac{ehp_f}{1000 \text{ sq ft}} \times 40.6$$

$$shp = ehp_{BH} \times 1.05 \div 0.62 \times 1.15 = ehp_{BH} \times 1.95$$

1.05 is allowance for appendages (rudder and bilge keels).
0.62 is assumed propulsive coefficient.
1.15 is allowance for sea power.

a 15 percent margin for sea power, is [ ] shaft horsepower.

This value of shaft horsepower is near enough to that used up to this stage (6800, page 162) so that no revision of machinery weight or fuel requirement is needed, and we are ready to draw the lines.

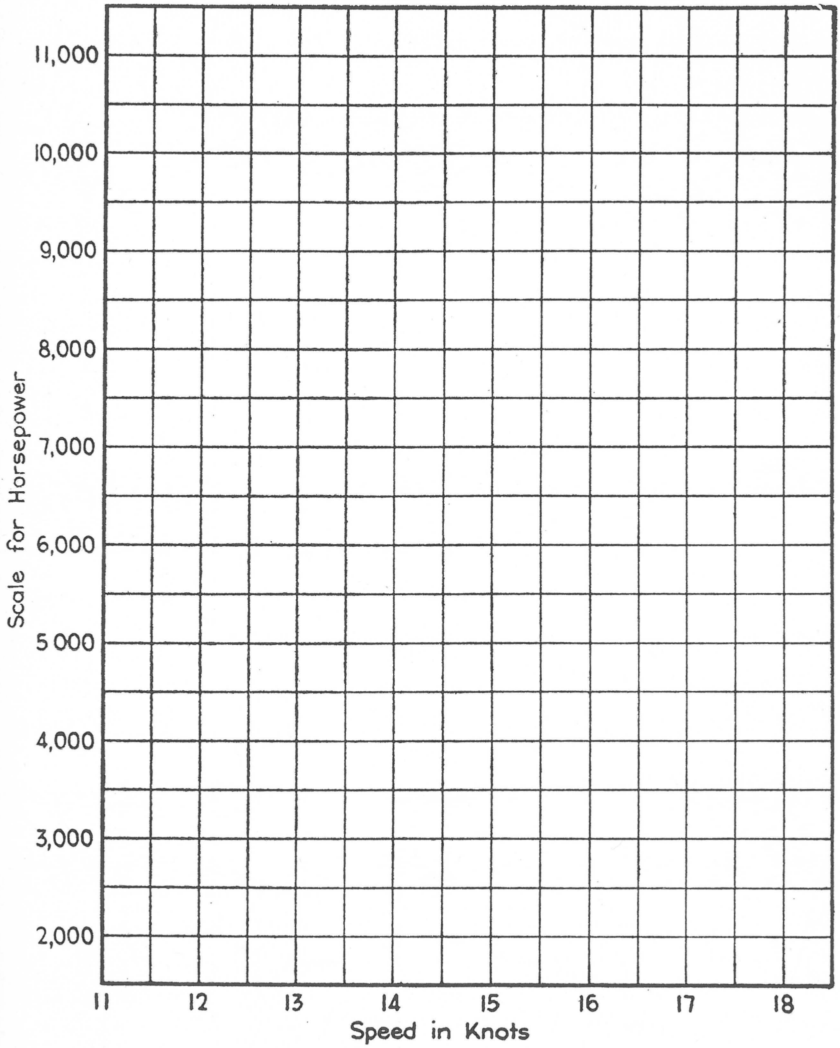


Fig. 60.—Horsepower curves for problem ship (by Taylor's standard series)

Section Area Curve

This important curve will control the lines of the ship. Its area must give the required prismatic coefficient, and its shape will have a pronounced effect on the resistance of the ship. Many model tank tests have been made exploring the effect on resistance of varying the shape of the section area curve, and much has been learned. For instance, a slow, full ship should have a hollow entrance in conjunction

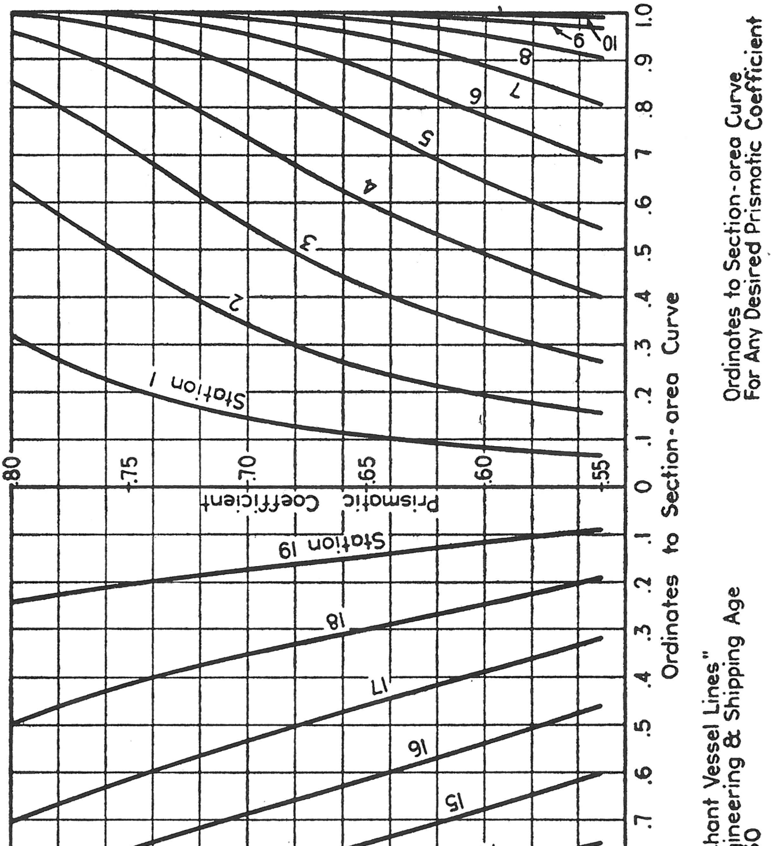


Fig. 61

with a full shoulder, while a fast ship should have a straight entrance, with as easy a shoulder as possible. In the after body, or "run," a hard shoulder is bad at all speeds.

For slow or moderate ships, a certain amount of parallel middle body reduces the resistance, as discussed in Taylor's "Speed and Power of Ships." The mid-point of this parallel middle body should be farther aft for a fast ship than for a slow ship. In practice a section area curve is either made similar to a past ship of similar speed-length ratio and prismatic coefficient which has proved successful, or by using some standardized data. Fig. 61, from the article on "Ship's Lines" by S. A. Vincent in *Marine Engineering* of March 1930, is an example of such data, and we will get our section area curve from it. The student is advised to read the whole article, since strictly the data should be used only in conjunction with the notes and discussion to be found in it.

Using Fig. 61 and Table 5, read and tabulate the ordinates of our section area curve to give the required prismatic coefficient of 0.728 (on the effective length). Plot these on Fig. 62 and draw a smooth curve through them to eliminate any unfairness, which if left unfaired would be passed on to the lines. Then test the faired values to see if they give the required prismatic coefficient by applying Simpson's rule, and modify slightly if necessary, keeping the curve fair. Then calculate our midship half-area as 29.5 feet $\times$ 27 feet $\times$ 0.978 = 779 square feet. Multiplying each faired section area ordinate by the midship half-area will give the half-area of each station, and we are ready for the freehand body plan. Table 5 (a), page 206, shows Table 5 filled in.

Lines

It is usual and convenient to draw "design lines" to a scale of ¼ inch to the foot, which for our ship would make a drawing about 10 feet long. For practice, however, ⅛-inch scale lines are just as good. The battens then need be only about 30 inches long, for with parallel middle body

the entrance and run may be faired separately. Tracing paper should be used, as it expands and contracts with humidity less than either heavy paper or tracing cloth. The procedure will be about as follows. The student is not expected to take time to fair the lines completely, but should if possible go through all the numbered steps below. Refer to the "Example Lines," Fig. 1, for guidance. The equipment required is a batten or spline of wood or celluloid for fairing lines, weights ("ducks") to hold the battens in place, and a planimeter for obtaining areas of stations.

1. Lay off the rectangular lines; that is, the vertical

TABLE 5.—ORDINATES FOR S.A. CURVE: 0.728 PRISMATIC

Sta.	S.A., from Fig. 65	S.A., faired	S.M.	S.A., for corrected prismatic		Half area, sq ft
0	0	0	× 1 =	0	0	0
1	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
2	[ ]	[ ]	× 2 =	[ ]	[ ]	[ ]
3	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
4	[ ]	[ ]	× 2 =	[ ]	[ ]	[ ]
5	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
6	[ ]	[ ]	× 2 =	[ ]	[ ]	[ ]
7	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
8	[ ]	[ ]	× 2 =	[ ]	[ ]	[ ]
9	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
10	1.000	1.000	× 2 =	2.000	1.000	779
11	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
12	[ ]	[ ]	× 2 =	[ ]	[ ]	[ ]
13	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
14	[ ]	[ ]	× 2 =	[ ]	[ ]	[ ]
15	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
16	[ ]	[ ]	× 2 =	[ ]	[ ]	[ ]
17	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
18	[ ]	[ ]	× 2 =	[ ]	[ ]	[ ]
19	[ ]	[ ]	× 4 =	[ ]	[ ]	[ ]
20	0	0	× 1 =	0	0	0
			Sum =	[ ]	[ ]	
Prismatic coefficient = sum ÷ 60			=	[ ]	0.728	

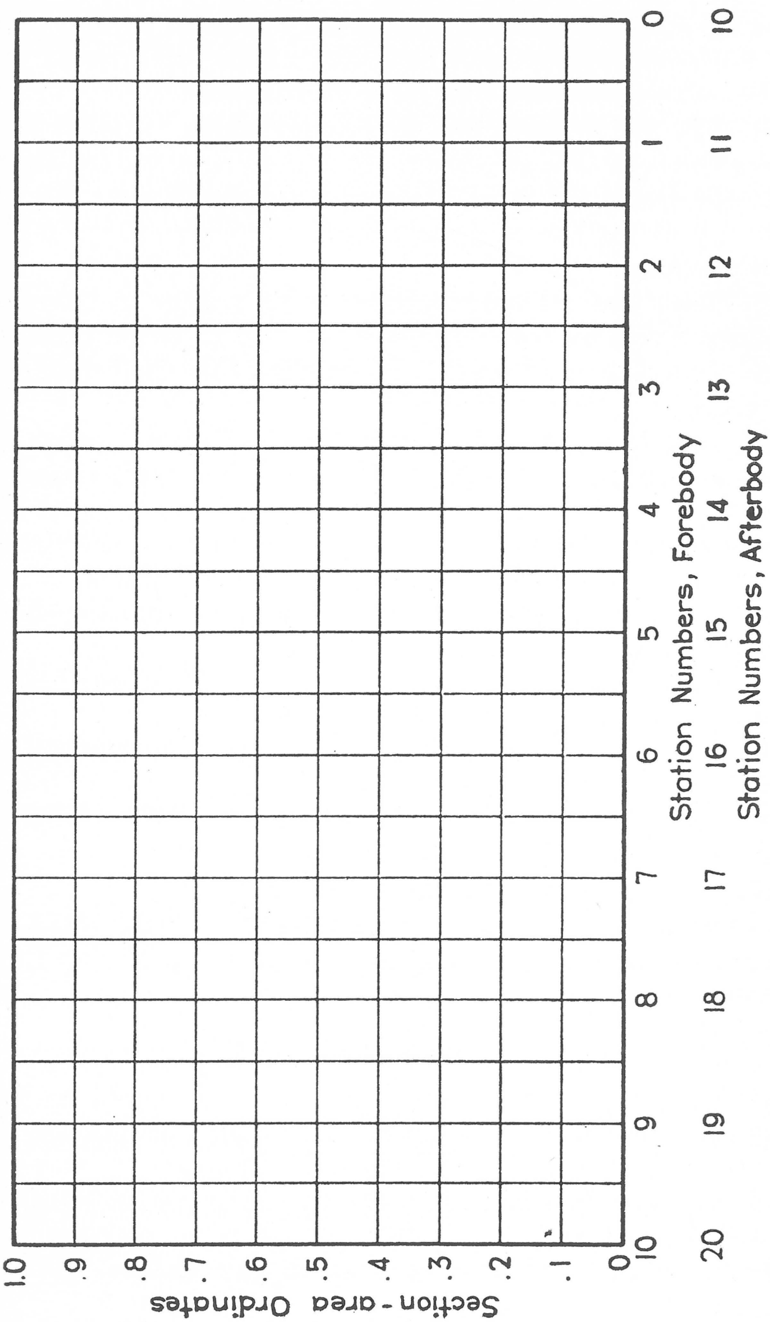


Fig. 62.—Section area curve for problem ship; prismatic coefficient = 0.728

lines for stations and the horizontal lines that will be used for both waterlines and buttocks. One line should be 27 feet above the base line for a load waterline.

2. Lay off a sheer line in profile, which is 34.25 feet above the base line at station 10, and using standard sheer from the freeboard rules, Reference (*a*) in Chapter XI.

3. Draw the bow and stern profile similar to those in the example lines.

4. Draw the midship section using a bilge radius and deadrise which by trial and error is found to give a midship section coefficient of 0.978. Note that this coefficient for the example lines was 0.983.

5. Sketch in, freehand, transverse sections similar to those shown on the example lines. (If we were drawing lines for a twin-screw ship, we would use a "broader-V" type of after sections.) Test each station with the planimeter and give it the required area from Table 5.

6. Get the height of deck at each station from the sheer line in No. 2 above.

7. Carry each station, freehand, up to the sheer height about as shown on the example lines.

8. Fair the load waterline, and change the stations to suit.

9. Fair the bilge diagonal, and change the stations to suit.

10. Fair a waterline at about half draft.

Continue as described in Chapter II. Care must be taken throughout the fairing not to depart appreciably from the required areas.

Curves of Form

When the lines are faired, curves of form would be prepared as described in Chapter II. There is nothing in this that has not been described in Chapter II, and the student is not expected to go through it unless he wants to do so for practice.

Data which have been omitted in this chapter for the student to fill in are as follows:

	Tons
Shell and framing	1243
Decks, etc.	684
Inner bottom	176
Main bulkheads	133
Total main hull (cubic number 0.285)	2652
Forecastle	34
Bridge	182
Poop	45
Total erections	261
Houses	233
Total houses, etc.	251
Total steel (cubic number coefficient 0.340)	3164
Inner bottom ceiling	49
Hold sparring	67
Sparring on bulkheads	16
Hatch covers	24
Carpenter, bridge deck	11
Carpenter, boat deck	11
Total carpenter work	203
Joiner house top	18
Accommodations	47
Total joiner work	101
Galley and pantry outfit	5
Bedding and linen	4
Furniture, etc.	12
Total steward's outfit	31
Equipment tonnage, under deck	6580
Erections	580
First tier houses	150
Second tier houses	110
Total	7420
Equipment numeral	39
Bower anchors, 9415 pounds	8.4
Spare anchors, 7980 pounds	3.6
Stream anchors, 3395 pounds	1.5
Chain, 300 fathoms, $2\frac{7}{16}$-inch, at 330 pounds	49.2
Total anchors, chains and lines	64.7
Wetted surface, square feet	40,800
Shaft horsepower	6700

Chapter XVI

SCANTLING PLANS, ARRANGEMENT PLANS, SPECIFICATIONS, FINAL DESIGN WEIGHT, CAPACITY PLAN, TONNAGE PLAN

We have now gone through the stages of design usually called "basic design," and discussed all of the subjects embraced by the term "theoretical naval architecture." The design of the problem ship has crystallized into definite dimensions, the general arrangement has been very roughly started, lines and curves of form have been drawn, and, unless we were unusually unfortunate in our approximations, any further changes in weight can be absorbed in a slight change of coefficients.

However, before the design can be turned over to the drawing room to be developed into working drawings, we must have

(a) General arrangement plans.
(b) Steel scantling plans.
(c) Specifications.

Also, the following design information must be prepared:

(d) Final design weight.
(e) Capacity plan of tanks and holds.
(f) Tonnage plan.
(g) Flooding data (if required).
(h) Longitudinal strength calculation (if required).
(i) Launching data (if required).

(a) and (c) are subjects which have not yet been discussed. The other items are all applications of what has been studied in preceding sections.

Items (a), (b) and (c) will necessarily proceed simultaneously, as each must be consistent with the other two. Items (d) to (i) depend on (a), (b) and (c) and while to a limited extent they can proceed simultaneously, they must in general wait until (a), (b) and (c) are done.

Preliminary general arrangement plans are the plans which show how the entire interior of the ship is utilized. They consist of a plan view of each deck and an inboard and outboard profile, all usually to a scale of $^1/_{16}$ inch to the foot. By these plans the designer makes sure that he has provided suitable and sufficient spaces for every requirement of the ship; cargo, machinery, crew, working spaces and passengers. The student who finds himself working on arrangement plans will have gotten into it gradually by tracing and filling in arrangements that have been developed by someone more experienced. This experience involves knowledge of the amount and character of spaces required for each phase of the ship's activity. Access, both regular and emergency, to all parts of the ship must be studied and provided. Passenger accommodations, both private and public, must conform to the standards of roominess and facilities prevailing on competing ships, so that the ship will not suffer by comparison with others. For instance, the size and character of passenger stateroom, which would be quite proper on our 460-foot freighter carrying 50 one-class passengers, would be far too small for a first-class stateroom on an express transatlantic liner.

Navigation laws regulate crew's spaces, and must be complied with. For instance, third-class passengers may be berthed three-high, but crew must not be berthed more than two-high.

Cargo-handling facilities must be provided which will permit efficient and rapid loading and unloading, and which will be adapted to the particular trade for which the ship is intended.

Subdivision, both watertight subdivision of the hull and fireproof subdivision throughout the hull and superstructure, is receiving steadily increasing attention from supervisory

authorities, and arrangements must be adapted to the limitations imposed by this subdivision.

We have touched on only a few of the problems which are worked out in the development of general arrangement plans.

Steel scantling plans. As has been described in Chapter V, a midship section plan will be drawn giving as many as possible of the steel scantlings which are to be used. For a small ship one drawing may be sufficient, but usually deck plans are needed to show such items as cannot be clearly or completely given on a midship section, such as pillars and girders, deck scantlings in way of openings, etc.

These steel plans and the general arrangement plans form the contract plans which together with the specifications (of which the contract plans are, strictly speaking, a part) form the basis of agreement between the owner and builder. They must, between the three of them, so completely describe and define the ship that they will answer all questions which will arise as to what is intended, and will leave no room for misunderstandings between the two parties.

In Chapter V the student had some experience in taking scantlings for the example ship out of the "Rules." He will not be expected to repeat the operation for the problem ship.

Specifications. Regarding these, as with arrangement plans, we can in this course give the student only a glimpse of what the designer must consider in writing them; we cannot give any practice at it.

Specifications will usually be divided into broad subdivisions about as follows:

A. *General.* Paragraphs describing the ship in a general way, the work to be done by the builders, work, if any, to be done by the owners, classification requirements, inspection and trials, etc.

B. *Hull Work.* This will be further subdivided into subjects similar to the divisions of hull weight on pages 76 and 77. Under steel, any scantlings which are shown on the steel scantling plans should not be repeated in the speci-

fications, for this leads to inconsistencies and contradictions. The equipment and furnishings of every space in the ship will be specified.

C. *Hull Engineering.* Paragraphs on each of the items shown on page 77.

D. *Propelling Machinery.* Boilers, engines, auxiliary machinery.

In general, information given on the contract plans will not be repeated in the specifications.

Standard Paragraphs. It would seem that standard paragraphs might be written up and filed on each item to be specified; for example, bitts, hatches, painting, tables, etc., etc., so that the writing of specifications would consist simply of selecting and arranging the pertinent paragraphs. However, although much certainly can be done in this direction, it is doubtful whether any naval architect has ever gotten his standard paragraphs into this ideal condition. For one thing, materials and processes are constantly developing and improving.

Final design weight. As soon as items (a), (b) and (c) above are sufficiently under way, the "final design weight" will be undertaken as described in Chapter VI. The steel estimate will be based on the actual lines and will use all the information shown on the steel scantling plans. The "wood and outfit" weights will follow the arrangement plans and specifications as closely as possible.

Unit weights will be carefully watched to see if they should be modified to suit what is actually specified. For instance, the joiner work unit weights of ten years ago would be badly misleading now, due to the development of new bulkheading materials and the increased use of fireproof construction. Welded bitts are far lighter than the old cast bitts. And so on.

If this weight calculation proves to be appreciably different from the preliminary estimate, the form coefficients must be changed to suit the revised weight, and the lines changed accordingly.

Capacity plan. A capacity plan will be made as soon as

the arrangement plans are sufficiently settled. The calculation of the capacities of the tanks and holds is described in Chapter IV. The capacity plan is a plan showing the results of these calculations. It consists usually of a hold plan, inner-bottom plan, and an inboard profile, to $^1/_{16}$ inch scale, with the volumes of each hold and tank either printed on the respective spaces, or more usually tabulated somewhere on the drawing. It will also show the total amount of cargo spaces, fuel, water and ballast tankage available.

Tonnage plan. A tonnage plan (see Chapter IV) will be made when the arrangement plans and scantling plans are settled. This is principally a body plan of the tonnage sections, drawn to the top of the inner bottom and inside of frames (or cargo battens, if fitted), together with sufficient dimensions and areas to enable the gross registered tonnage to be calculated from the plan.

Floodable-length drawing. The floodable-length drawing, if one is required, will show the results of the floodable-length calculation described in Chapter XI. It will show an inboard profile, on which the main subdivision bulkheads are drawn especially prominently. The curves of floodable length will be drawn on this profile, together with the sloping lines showing that the compartmentation is satisfactory. There will also be a hold plan on the drawing, showing any steps or recesses in the bulkheads, and any thwartship subdivision that affects the subdivision. A table of pertinent data, including principal dimensions, permeabilities, volumes, etc., will complete the drawing.

Longitudinal strength drawing. The longitudinal strength drawing, if one is required, will show figures (e), (f) and (g) of Fig. 34, usually superimposed on a common base line for compactness. There will be two diagrams for the two conditions of loading usually used for hogging and sagging conditions, respectively. Tables of weight groups, total weight, and resulting maximum stresses in deck and keel, will be shown.

Launching drawing. A launching drawing similar to Fig. 51 will be made as described in Chapter XII.

The naval architect's office is now through with the ship until it is built. Then there will be two jobs still to do.

Inclining experiment. An inclining experiment (see Chapter III) will be performed, if the ship is to carry passengers, and the finished "light ship" vertical and horizontal centers of gravity determined. With this "light ship" as a starting point, various conditions of loading, such as

leaving port, full cargo, full fuel and water,
arriving port, full cargo, no fuel and water,
leaving port, no cargo, full fuel and water,
arriving port, no cargo, no fuel and water,

and the *GM* in each condition calculated. If water ballast is required to obtain sufficient *GM*, the amount and location are found. This information is furnished to the owners, that they may be guided thereby in operating the ship.

Finished weights will, if time permits, be calculated from the working drawings for the ship, as discussed in Chapter VI, to guide the estimator of weights of future similar ships.

This concludes our course on theoretical naval architecture. If the student finds himself with time available, there is no limit to the work he may do, if he wishes, in finishing off the design of the problem ship. A few suggestions are given below. These jobs will not explore any new ground; but, if seriously undertaken, they will give a firmer grasp on the various subjects.

1. Draw an outline midship section of the problem ship and obtain scantlings from the American Bureau of Shipping rules as was done for the example ship in Chapter V.

2. Make up a possible crew list for this ship, from published arrangement plans of ships similar to our problem ship.

3. Assume the boiler and engine room flooded, when the ship is loaded to a 29-foot draft, displacement 14,740 tons, *GM* +3.6 feet. Following the steps outlined in Chapter XI, but neglecting trim, find the flooded draft and *GM*. Neglect also the slight variation of transverse area of the

ship in the boiler and engine rooms; that is, assume that they are in the parallel middle body. The permeability of the machinery spaces could be taken as 0.80, and of the waterplane as 0.95.

(... Ship) To 24'-0" W.L.

$\ldots 2 \times L.I. =$ Trans. Mom. of Inertia

$\frac{\ldots\text{of Inertia}}{\ldots\text{. Displ.}}$ = Trans. B.M.

TRANS. B. M.

Column No.		(3)	(12)
	W.^{3}	S.M.	Products

...n$_6$ =

$\frac{\text{Sum}_6 \times 2 \times L.I.}{9} = \underline{\quad} \times \frac{2}{9} \times \underline{\quad}$

SUMMARY		
Molded Displacement S.W.		
Shell & Appendg. Displ. (6%)		*
Total S.W. Displacement		⊕
Block Coeff.		⊕
Max. Section Coeff.		⊕
Prismatic Coeff.		⊕
Water Plane Coeff.		⊕
Tons per Inch S.W.		⊕
C.G. of Water Plane __ of ⊠		⊕
Trans. B. M.		⊕
C.B. Above Base Line		⊕ *
Trans. "M" Above Base Line		⊕
C.G. Above Base Line	17.10	*
Trans. "G.M." Full Load Condition		
Longtl. "B.M."		
C.B. Above Base Line		
Longtl. "M" Above Base Line		
C.G. Above Base Line	17.10	
Longtl. "G.M." Full Load Condition		
Foot Tons M to Trim 1"		⊕
Longtl. C.B. __ of ⊠		⊕
Incr. in Displt. per Foot Trim by Stern		⊕

$= \frac{}{\text{(Trans. Mom. of Inertia)}} \div \frac{}{\text{(Cu. Ft. Displ.)}} = \underline{\text{Trans. B. M.}}$

$\frac{}{\ldots\text{u. Ft. Displ.)}} = \underline{\text{Longl. B. M.}}$

* Calculations made on Separate Sheet

⊕ Usually Plotted on "Curves of Form"

$\ldots\text{rn} = \frac{\text{Tons per Inch} \times 12 \times \text{C.G. of W.P. Aft of } ⊠}{\text{L.W.L.}}$

$= \frac{\times 12 \times}{415} =$

$\frac{\times}{12 \times} =$

...s Formula

Effective Length 415'-0"

Curves of Form Cal

Longl. Interval = $\frac{\text{Effective Length}}{20}$ = ____

$\frac{\text{L.I.} \times 2}{3}$ = Multiplier

$\text{Sum}_1 \times$ Multipr. = Cu. Ft. Mld. Displ.

Sum_2 = Diff. of Mom.

$\frac{\text{Sum}_2}{\text{Sum}_1} \times$ L.I. = L.C.B

$\frac{\text{Sum}_3 \times 2 \times \text{L.I.}}{3}$ = Area W.P.

$\frac{\text{Area of W.P.}}{420}$ = T.P.I.

$\frac{\text{Sum}_4 \times}{\text{Sum}}$

$\frac{\text{Sum}_5 \times}{3}$

Mom. of

Column No.	DISPLACEMENT				LONGL. C. B.		TONS PER INCH				
	①	②	③	④	⑤	⑥	⑦	③	⑧	⑤	
	Sta.	Plan't'r Reading	Half Area	S.M.	Products	L.L.	Moments	½ Breadth of W.L.	S.M.	Products	L.L.	H.I
	0	000						00.0				
	1	155						8.2				
	2	342						16.5				
	3	493						22.6				
	4	574						25.5				
	5	612						26.6				
	6	626						27.0				
	7	626						27.0				
	8	626						27.0				
	9	626						27.0				
	10	626				Ford.		27.0			Ford.	
	11	626						27.0				
	12	626						27.0				
	13	618						27.0				
	14	602						27.0				
	15	565						27.0				
	16	503						26.8				
	17	414						26.0				
	18	289						22.7				
	19	139						17.3				
	20	000						00.0				
		Sum_1 =				Aft		Sum_3 =			Aft	

X Multiplier ____

= ____ Cu. Ft. Displ.

÷ 35

= ____ S.W. Tons Displ.

Sum_2

$\frac{\text{Sum}_2 \times \text{L.I.}}{\text{Sum}_1}$ = ____ x ____

= ____ Longl. C.B.

____ of Ⅲ

Ford or Aft

$\frac{\text{Sum}_3 \times 2 \times \text{L.I.}}{3}$ = ____ (Area W.P.)

$\frac{\text{Area of W.P.}}{420}$ = ____ (Tons per Inch)

Sum_4

$\frac{\text{Sum}_4 \times \text{L.I.}}{\text{Sum}_3}$ =

=

(Ford or Aft

Plan'r. Constant = $\frac{\text{Half Beam} \times \text{Draft}}{\text{Reading for Ⅲ Half Rect.}} = \frac{27 \times 24}{636}$ = 1.0176

$\frac{\text{Area W.P}}{\text{B.} \times \text{L.}}$ = Water Plane Coeff. =

Block Coeff = $\frac{\text{Cu. Ft. Displ.}}{\text{L.} \times \text{B.} \times \text{d.}}$ = $\frac{\quad}{\quad \times \quad \times \quad}$ = 0.____

Increase in D

(If C.G. of W.P

(Substitute "

Max. Section Coeff. = $\frac{2 \times \text{Max. Half Area}}{\text{B.} \times \text{d.}}$ = $\frac{2 \times \quad}{\quad \times \quad}$ = 0.____

Prismatic Coeff. = $\frac{\text{Block Coeff.}}{\text{Max. Sec. Coeff}}$ = $\frac{0.}{0.}$ = 0.____

! Mom. to Trir

! B.M. is Usual

Fig. 4.—Curves

...culations for (Example Ship) To 24'-0" W. L.

$\frac{\ldots L.I.}{3} =$ C.G. of W.P. Abt. ⊠

$\frac{\ldots L.I.^3 \times 2}{3} =$ Mom. of Inertia of W.P. Abt. ⊠

$\frac{\ldots I - (\text{Area W.L.} \times \text{C.G.})^2}{\text{Cu. Ft. Displ.}} =$ Longl. B. M.

$\frac{\text{Sum}_6 \times 2 \times L.I.}{9} =$ Trans. Mom. of Inertia

$\frac{\text{Mom. of Inertia}}{\text{Cu. Ft. Displ.}} =$ Trans. B. M.

LONGL. B. M.			TRANS. B. M.		
(9)	(5)	(10)	(11)	(3)	(12)
B. x L. L.	L.L.	H.B. x (L.L.)^2	$\frac{1}{2}$ B. of W.^{3}	S.M.	Products
	Sum_5		$\text{Sum}_6 =$		

SUMMARY		
Molded Displacement S.W.		
Shell & Appendg. Displ. (6%)		#
Total S.W. Displacement		⊕
Block Coeff.		⊕
Max. Section Coeff.		⊕
Prismatic Coeff.		⊕
Water Plane Coeff.		⊕
Tons per Inch S.W.		⊕
C.G. of Water Plane __ of ⊠		⊕
Trans. B. M.		
C.B. Above Base Line		⊕ #
Trans. "M" Above Base Line		⊕
C.G. Above Base Line	17.10	#
Trans. "G.M." Full Load Condition		
Longtl. "B.M."		
C.B. Above Base Line		
Longtl. "M" Above Base Line		
C.G. Above Base Line	17.10	
Longtl. "G.M." Full Load Condition		
Foot Tons M to Trim 1"		⊕
Longtl. C.B. __ of ⊠		⊕
Incr. in Displt. per Foot Trim by Stern		⊕

x

C.G. of W.P.

__ of ⊠

$\frac{\text{Sum}_5 \times 2 \times L.I.^3}{3} = \_\_ \times \frac{2}{3} \times \_\_$

$= \frac{\qquad}{\text{(Mom. of Inertia Abt. ⊠)}}$

$- \frac{\qquad}{(\text{Area W.P.} \times \text{C.G.}^2)}$

$= \frac{\qquad}{\text{(Mom. of Inertia Abt. C.G. of W.P.)}} \div \frac{\qquad}{\text{(Cu. Ft. Displ.)}} = \underline{\quad\text{Longl. B. M.}\quad}$

$\frac{\text{Sum}_6 \times 2 \times L.I.}{9} = \_\_ \times \frac{2}{9} \times \_\_$

$= \frac{\qquad}{\text{(Trans. Mom. of Inertia)}} \div \frac{\qquad}{\text{(Cu. Ft. Displ.)}} = \underline{\quad\text{Trans. B. M.}\quad}$

__ x __ = 0.

Calculations made on Separate Sheet

⊕ Usually Plotted on "Curves of Form"

...Displacement per Foot of Trim by Stern = $\frac{\text{Tons per Inch} \times 12 \times \text{C.G. of W.P. Aft of ⊠}}{\text{L.W.L.}}$

...P. should be Forward of Midships)
...Decrease " for Increase

$= \frac{\_\_ \times 12 \times \_\_}{415} =$

...n 1" = $\frac{\text{Tons Displ.} \times \text{Longl. G. M.}}{12 \times \text{L.W.L.}} = \frac{\_\_ \times \_\_}{12 \times \_\_} =$

...ly Substituted for G.M. in this Formula

...s of form calculations

ANSWERS TO PROBLEMS

Chapter II

Problem 1. See Fig. 4(a).
Problem 2. See Fig. 7(a).
Problem 3. (a) 21.7 feet. (b) 2.55 feet. (c) 9 inches by the head.
Problem 4. 20 feet 5.7 inches.
Problem 5. 16 feet for triangle, 10 feet for rectangle; difference 6 feet.
Problem 6. Yes. Drafts with forepeak filled: (about) 11 feet 7½ inches forward, 11 feet 8¾ inches aft.

Chapter III

Problem 1. (a) 3.75 feet. (b) 3 degrees, nearly.
Problem 2. 0.84 foot.
Problem 3.

10°	20°	30°	40°	50°
0.67′	1.48′	2.47′	3.16′	3.13′

Problem 4.

Lowered, inches	Displacement, tons	Virtual *CG*, feet	*KM*, feet	*GM*, foot
12	10,050	21.3	22.6	+0.7
36	9000	23.8	22.8	−1.0

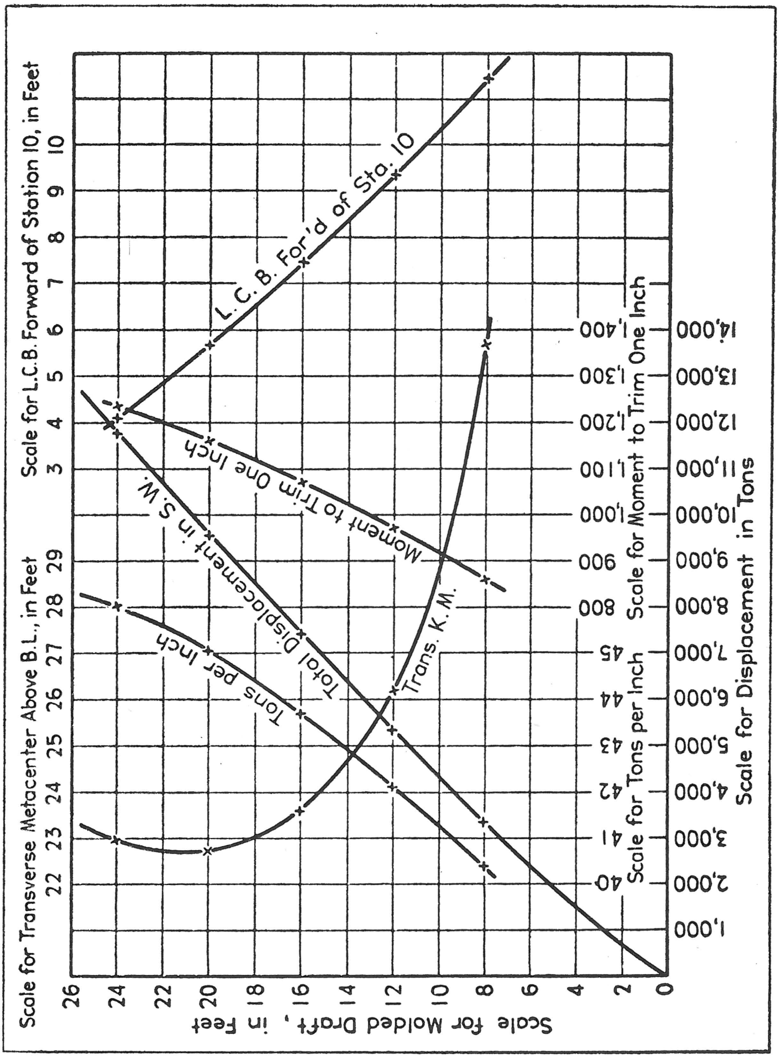


Fig. 7 (a).—Curves of form for example ship

CHAPTER IV

Problem 1.

Sounding	Tons	Sounding	Tons	Sounding	Tons	Sounding	Tons	Sounding	Tons	Sounding	Tons
0′0″	0	0′6″	0.1	1′0″	0.3	1′6″	0.5	2′0″	0.8	2′6″	1.1
3′0″	1.5	3′6″	2.0	4′0″	2.6	4′6″	3.3	5′0″	4.0	5′6″	4.8
6′0″	5.7	6′6″	6.6	7′0″	7.5	7′6″	8.5	8′0″	9.4	8′6″	10.4
9′0″	11.3	9′6″	12.3	10′0″	13.2	10′6″	14.2	11′0″	15.1	11′6″	16.1
12′0″	17.0	12′6″	17.9	13′0″	18.9	13′6″	19.8	14′0″	20.8	14′6″	21.7
15′0″	22.7	15′6″	23.6	16′0″	24.6	16′6″	25.5	17′0″	26.5	17′4″	27.1

Problem 2. 5.78 feet above bottom of tank.

Problem 3.

Under and 'tween deck	5870	Propelling power	1960
Poop, bridge and forecastle	60	Crew spaces	160
Houses and excess hatches	120	Working spaces	50
Light and air space	90		——
	——	Deductions	2170
Total gross	6140		
Net	3970		

CHAPTER V

Problem 1. The 6-inch channel; stress is 14,800 pounds per square inch.

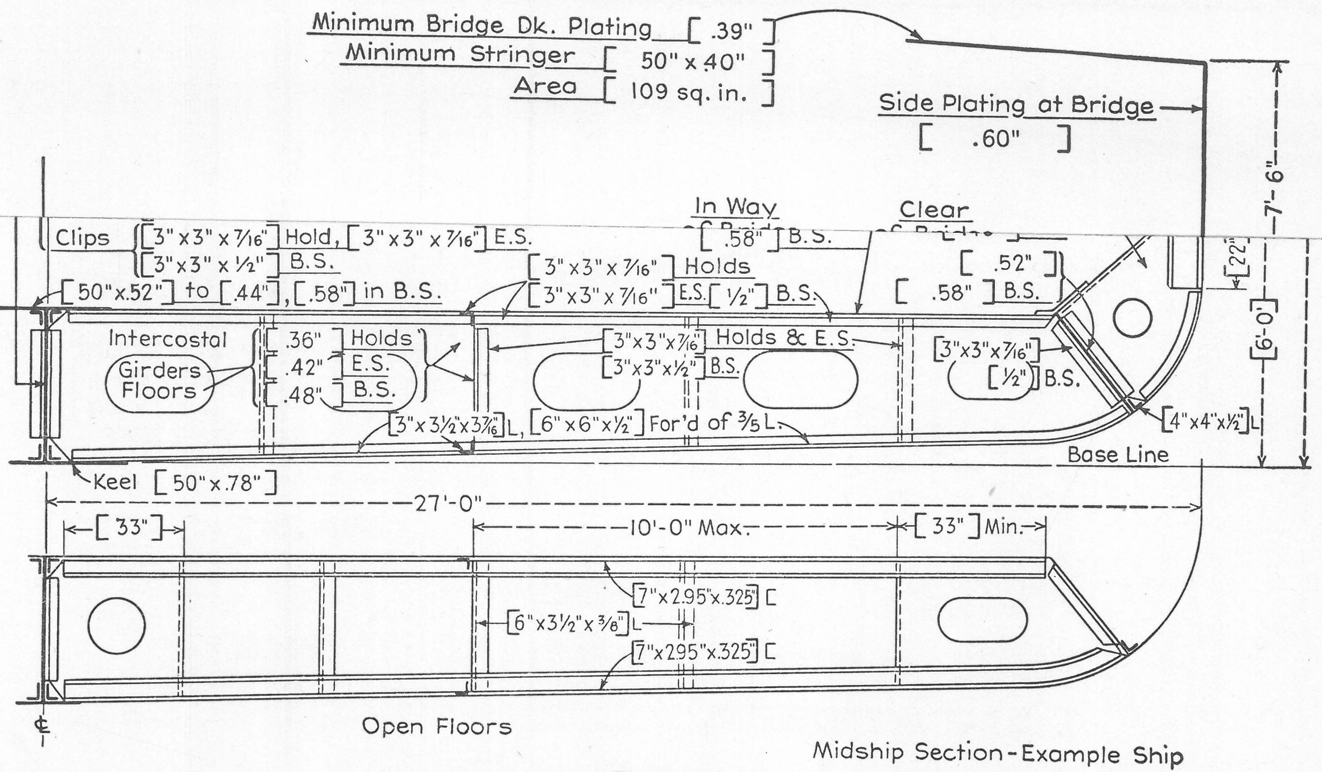
Problem 2. 3½-inch diameter, actual stress 7800 pounds per square inch.
Permissible stress 8800 pounds per square inch.

Problem 3. Tearing of plate; efficiency 66 percent.

Problem 4. 8½ inches (10,000 pounds per square inch = 3535 pounds per inch of weld).

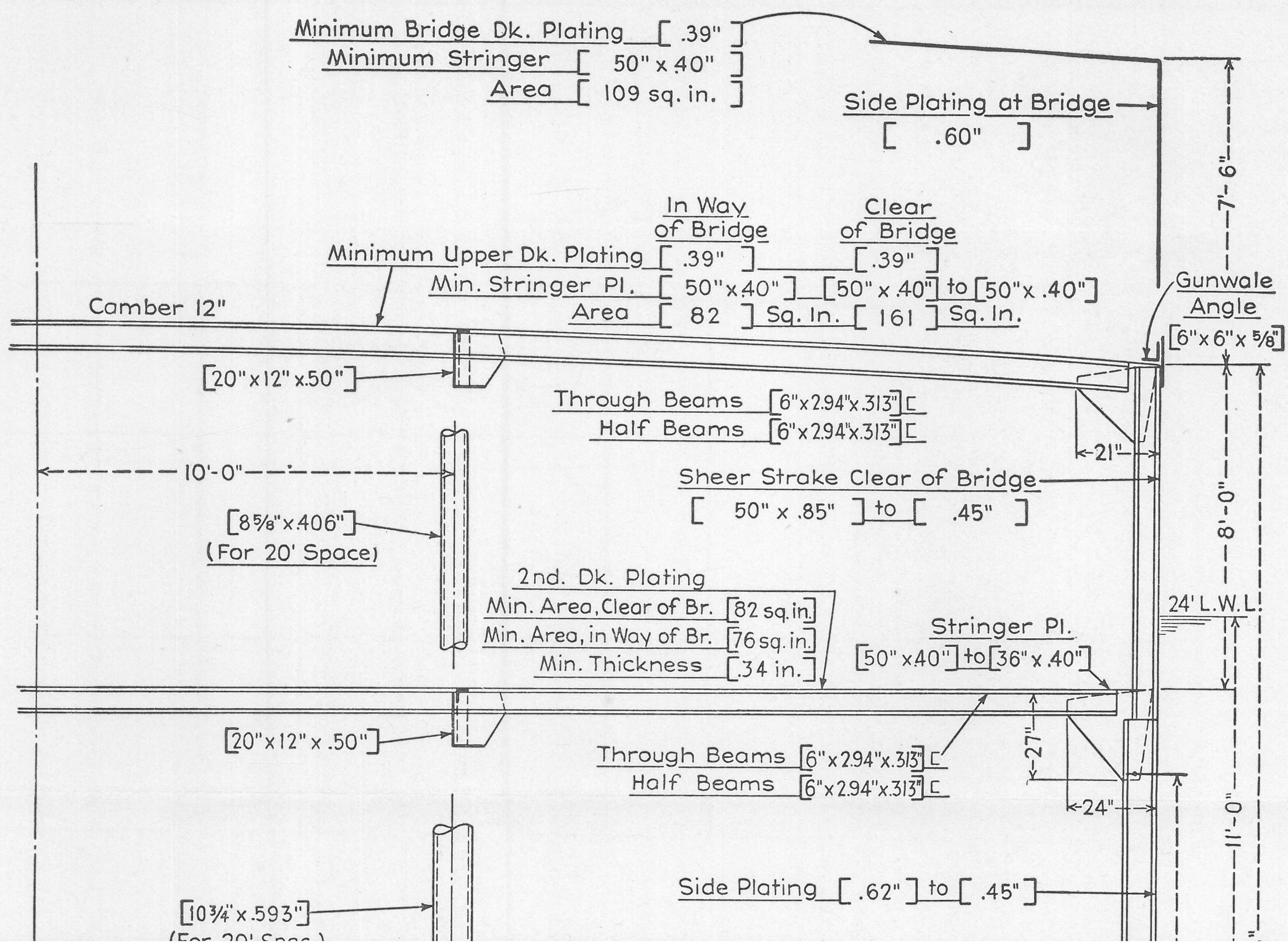
Item	Section	Table	Size
Rule dimensions	2	. . .	420′ × 54′ × 30′3″ × 24′0″
Frame spacing	8(2)	4 and 13	30″ to 24″ at ends
Flat plate keel	4(1)	13	0.50″ × 0.78″
Center girder	7(3a)	3 and 4	42″ × 0.52″ to 0.44″, 0.58″ B.S.
Bottom angles	7(3c)	3	4″ × 4″ × ½″, Dbl.

Item	Section	Table	Size
Top angles	7(3b)	4	$3\frac{1}{2}'' \times 3\frac{1}{2}'' \times \frac{7}{16}''$, $\frac{1}{2}''$ in B.S.
Solid floor—Plates	7(4)	4	0.36″ Holds, 0.42″ E.S., 0.48″ B.S.
Spacing—under boiler and engine bearers	7(4)	...	Every frame
Spacing in forward $\frac{1}{5}$ length	7(4)	...	Every frame
Spacing elsewhere	7(4)	...	Every 4th frame
Frame bars	7(4f)	4	$3'' \times 3\frac{1}{2}'' \times \frac{7}{16}''$, $6'' \times 6'' \times \frac{1}{2}''$ in forward $\frac{1}{5}$ L
Reverse frames	7(4h)	4	$3'' \times 3'' \times \frac{7}{16}''$, $\frac{1}{2}''$ B.S. Dbl. under E. & B.
Stiffeners	7(4d)	4	Same as reverse frames
Clips to center girder	7(4e)	25K	$3'' \times 3'' \times \frac{7}{16}''$, $\frac{1}{2}''$ B.S. Dbl. under bearers
Clips to margin	...	25K	$3'' \times 3'' \times \frac{7}{16}''$, $\frac{1}{2}''$ B.S.
Clips to intercostal girder	7(8g)	25K	$3'' \times 3'' \times \frac{7}{16}''$, $\frac{1}{2}''$ B.S.
Open floors—Brackets	7(5c)	...	Same as solid floors
Over-lap on brackets	7(5c)	...	$0.05B = 33''$
"W"	7(5c)	...	$2.5 \times 24 \times 5 \times 0.03 = 9.0$
Strut	7(5c)	...	$6'' \times 3\frac{1}{2}'' \times \frac{3}{8}''$ angle
"N"	7(5b)	...	$\frac{1}{2} \times 2.5 \times 24 = 30$
"I"	7(5b)	...	10
Frame and Reverse	7(5b)	5	$7'' \times 2.95'' \times 0.325''$ channel
Inner bottom—Centerline	7(6b)	4	$50'' \times 0.52''$ to 0.44″, 0.58″ B.S.
Margin	7(6e)	4	$32'' \times 0.52''$, 0.58″ B.S.
Remainder	7(6a)	4	0.45″ Holds, 0.52″ E.S., 0.58″ B.S.
Margin angle	7(6f)	4	$4'' \times 4'' \times \frac{1}{2}''$
Intercostal girder	7(8)	4	Same as solid floors
Top angles	7(8f)	3 notes	Same as reverse frames on solid floors
Bottom angles	7(8e)	3 notes	Same as frames on solid floors



Minimum Bridge Dk. Plating [.39"]
Minimum Stringer [50" x 40"]
Area [109 sq. in.]
Side Plating at Bridge
[.60"]
7'- 6"
In Way
[.58"] B.S.
Clear
[.52"]
[.58"] B.S.
[22"]
[6'-0']
Clips [3" x 3" x 7/16"] Hold, [3" x 3" x 7/16"] E.S.
[3" x 3" x 1/2"] B.S.
[50" x .52"] to [.44"], [.58"] in B.S.
[3" x 3" x 7/16"] Holds
[3" x 3" x 7/16"] E.S. [1/2"] B.S.
Intercostal
Girders
Floors
[.36"] Holds
[.42"] E.S.
[.48"] B.S.
[3"x3"x7/16"] Holds & E.S.
[3"x3"x1/2"] B.S.
[3"x3"x7/16"]
[1/2"] B.S.
[3" x 3 1/2" x 3 7/16"] L, [6" x 6" x 1/2"] For'd of 3/5 L.
[4"x4"x1/2"] L
Base Line
Keel [50" x .78"]
27'-0"
[33"]
10'-0" Max.
[33"] Min.
[7" x 2.95" x .325"] C
[6" x 3 1/2" x 3/8"] L
[7" x 2.95" x .325"] C
Open Floors
Midship Section - Example Ship

Fig. 28 (a)


Minimum Bridge Dk. Plating [.39"]
Minimum Stringer [50" x .40"]
Area [109 sq. in.]
Side Plating at Bridge
[.60"]
7'-6"
In Way of Bridge
Clear of Bridge
Minimum Upper Dk. Plating [.39"] [.39"]
Min. Stringer Pl. [50" x .40"] [50" x .40"] to [50" x .40"]
Area [82] Sq. In. [161] Sq. In.
Camber 12"
Gunwale Angle
[6" x 6" x 5/8"]
[20" x 12" x .50"]
Through Beams [6" x 2.94" x .313"]
Half Beams [6" x 2.94" x .313"]
21"
10'-0"
Sheer Strake Clear of Bridge
[50" x .85"] to [.45"]
8'-0"
[8 5/8" x .406"]
(For 20' Space)
2nd. Dk. Plating
Min. Area, Clear of Br. [82 sq. in.]
Min. Area, in Way of Br. [76 sq. in.]
Min. Thickness [.34 in.]
24' L.W.L.
Stringer Pl.
[50" x .40"] to [36" x .40"]
[20" x 12" x .50"]
27"
Through Beams [6" x 2.94" x .313"]
Half Beams [6" x 2.94" x .313"]
24"
11'-0"
Side Plating [.62"] to [.45"]
[10 3/4" x .593"]

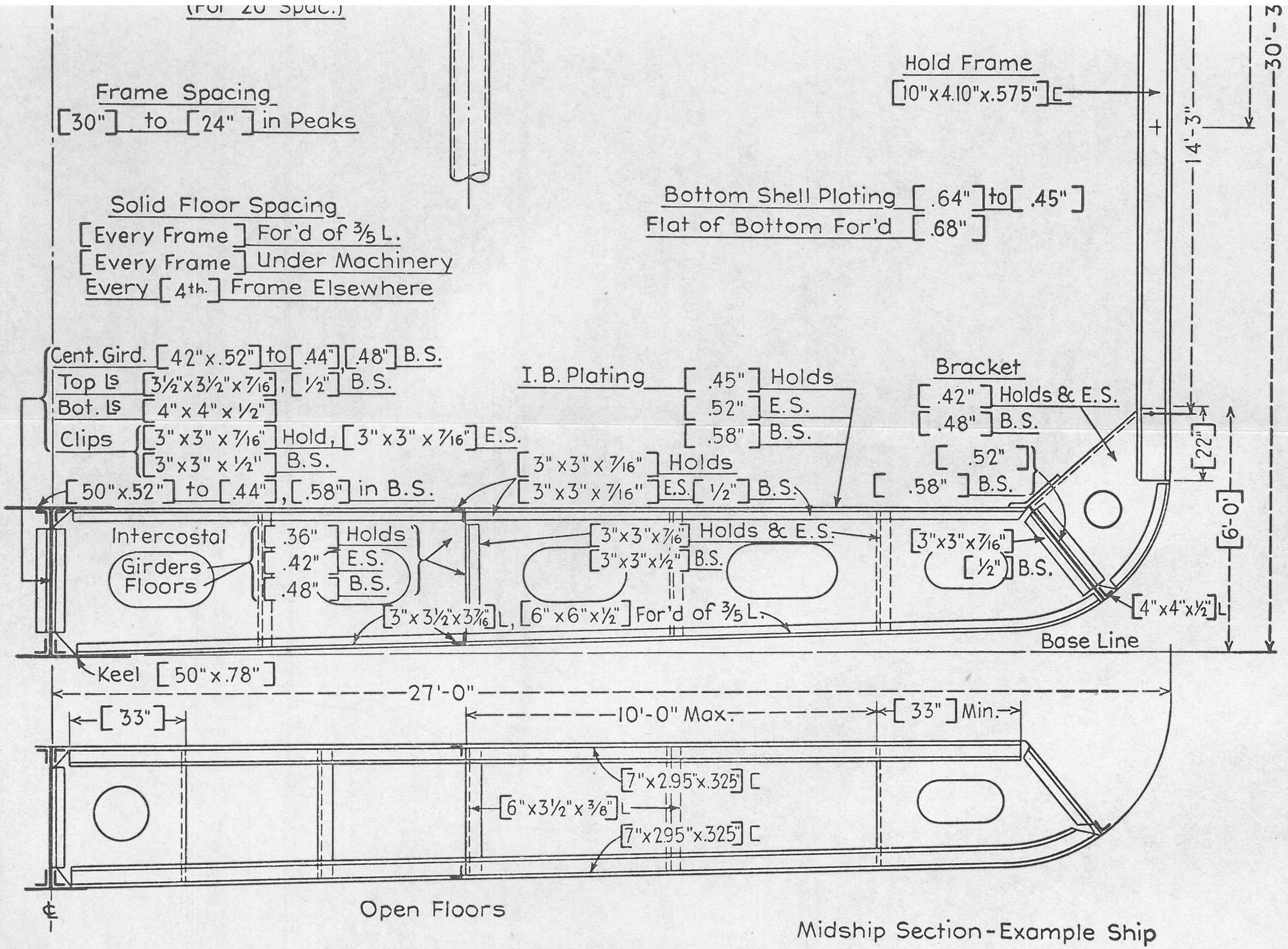
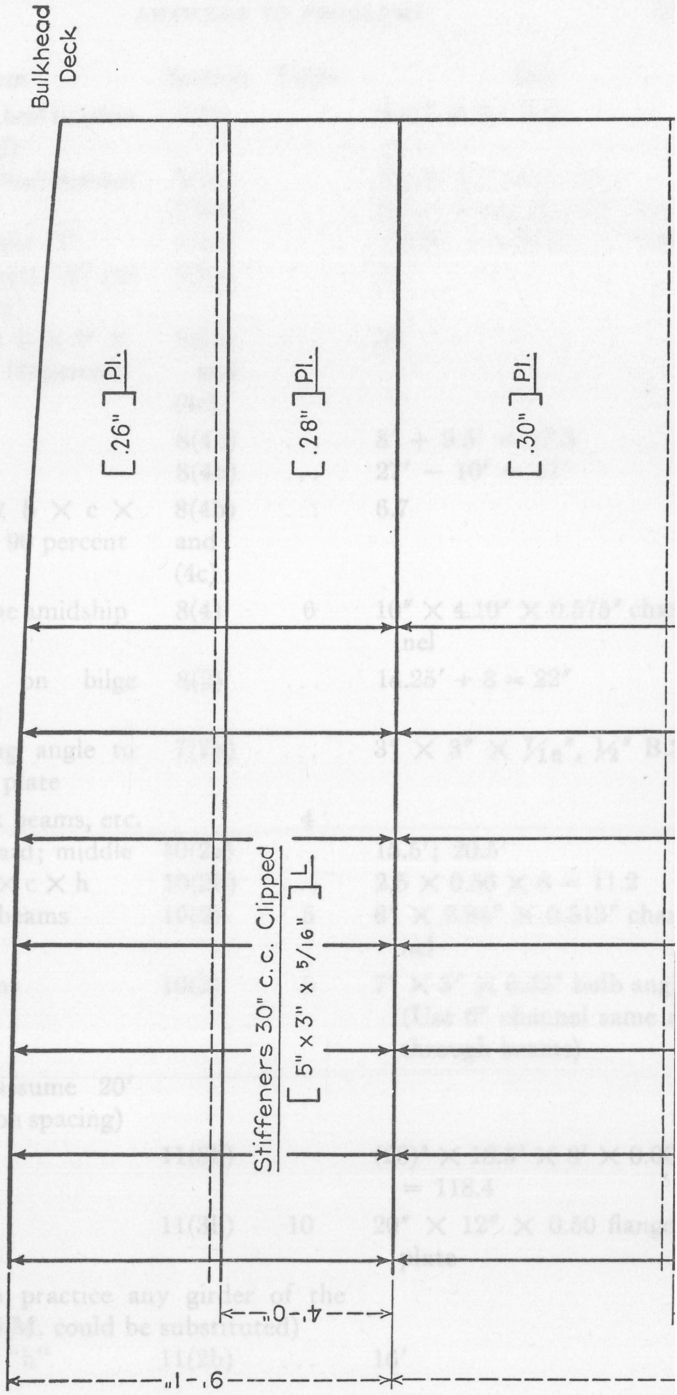

(For 20 Spac.)
Frame Spacing
[30"] to [24"] in Peaks
Solid Floor Spacing
[Every Frame] For'd of 3/5 L.
[Every Frame] Under Machinery
Every [4th.] Frame Elsewhere
Hold Frame
[10"x4.10"x.575"] C
14'-3"
30'-3
Bottom Shell Plating [.64"] to [.45"]
Flat of Bottom For'd [.68"]
Cent. Gird. [42"x.52"] to [.44"], [.48"] B.S.
Top Ls [3½"x3½"x7/16"], [½"] B.S.
Bot. Ls [4"x4"x½"]
Clips [3"x3"x7/16"] Hold, [3"x3"x7/16"] E.S.
[3"x3"x½"] B.S.
[50"x.52"] to [.44"], [.58"] in B.S.
I. B. Plating [.45"] Holds
[.52"] E.S.
[.58"] B.S.
[3"x3"x7/16"] Holds
[3"x3"x7/16"] E.S. [½"] B.S.
Bracket
[.42"] Holds & E.S.
[.48"] B.S.
[.52"]
[.58"] B.S.
[22"]
[6'-0"]
Intercostal
Girders
Floors
[.36"] Holds
[.42"] E.S.
[.48"] B.S.
[3"x3"x7/16"] Holds & E.S.
[3"x3"x½"] B.S.
[3"x3"x7/16"]
[½"] B.S.
[4"x4"x½"] L
[3"x3½"x37/16"] L, [6"x6"x½"] For'd of 3/5 L.
Base Line
Keel [50"x.78"]
27'-0"
[33"]
10'-0" Max.
[33"] Min.
[7"x2.95"x.325"] C
[6"x3½"x3/8"] L
[7"x2.95"x.325"] C
Open Floors
℄
Midship Section - Example Ship

Fig. 28 (a)


Bulkhead Deck
.26" Pl.
.28" Pl.
.30" Pl.
Stiffeners 30" c.c. Clipped
5" x 3" x 5/16" L
4'-0"
9'-1"

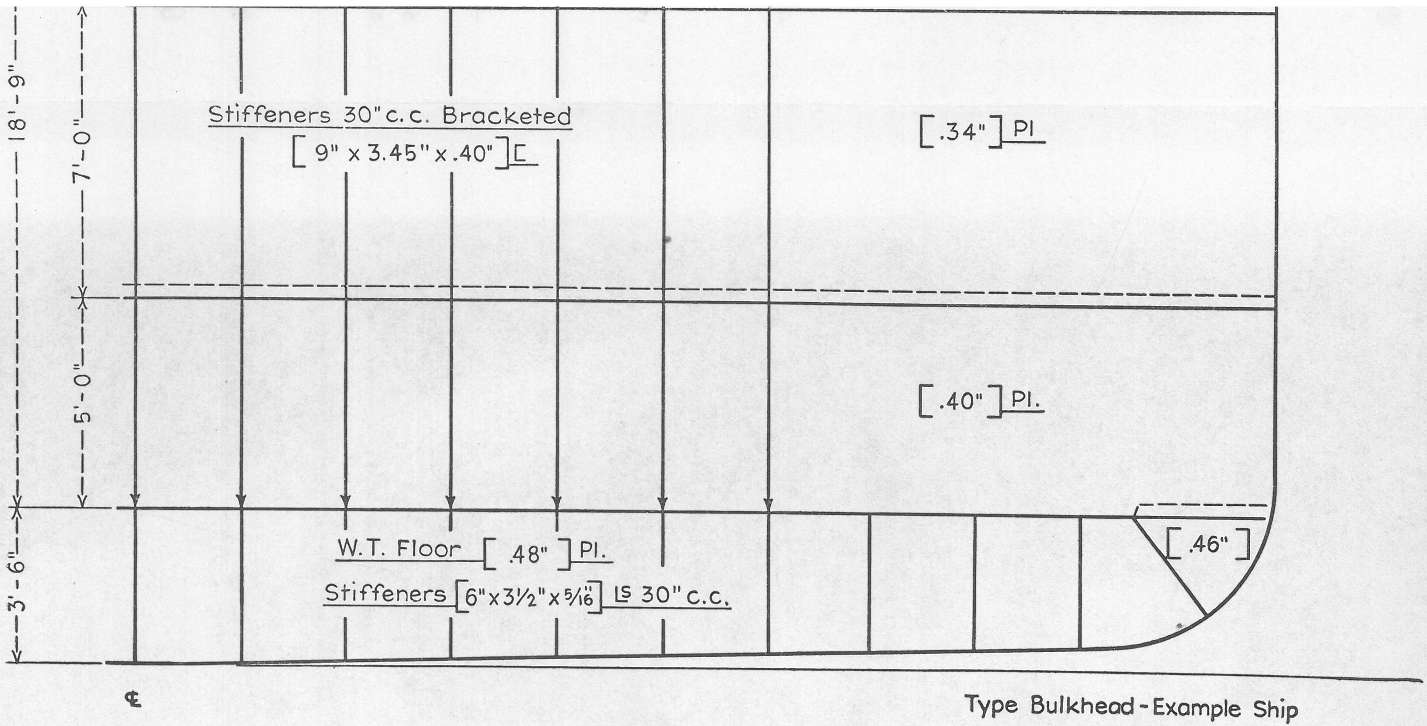

18'-9"
7'-0"
5'-0"
3'-6"
Stiffeners 30" c.c. Bracketed
[9" x 3.45" x .40"] [
[.34"] Pl
[.40"] Pl.
W.T. Floor [.48"] Pl.
Stiffeners [6" x 3½" x 5/16"] Ls 30" c.c.
[.46"]
℄
Type Bulkhead - Example Ship

Fig. 29 (a)

Item	Section	Table	Size
Hold frame, heel bracket (flanged)	7(7)	...	0.42″, 0.48″ B.S.
Height of heel bracket	8(3)	...	Made 6′, nearly 2D.
Gusset	7(7c)	...	Every frame, five ⅞″ rivets
Approximate "l"	8(4a)	...	(14.25′ + 0.002L) = 15.09′
Approximate "h" (to load line)	8(4a)	...	11′
$M = s \times h \times l^2 \times 0.01 \times 90$ percent	8(4a) and (4c)	...	56
c	8(4b)	...	8′ + 9.5′ = 17.5′
b	8(4b)	...	27′ − 10′ = 17′
$K = s \times b \times c \times 0.01 \times 90$ percent	8(4b) and (4c)	...	6.7
Hold frame amidship	8(4)	6	10″ × 4.10″ × 0.575″ channel
Over-lap on bilge bracket	8(3)	...	14.25′ ÷ 8 = 22″
Connecting angle to margin plate	7(7b)	...	3″ × 3″ × 7⁄16″, ½″ B.S.
Second deck beams, etc.		4	
"l" outboard; middle	10(2a)	...	15.5′; 20.5′
"N" = $s \times c \times h$	10(2b)	...	2.5 × 0.56 × 8 = 11.2
Through beams	10(2)	5	6″ × 2.94″ × 0.313″ channel
Half beams	10(2)	5	7″ × 3″ × 0.32″ bulb angle (Use 6″ channel same as through beams)
Girder (assume 20′ stanchion spacing)			
"M"	11(3b)		$(20)^2$ × 18.5′ × 8′ × 0.002 = 118.4
Girder	11(3b)	10	20″ × 12″ × 0.50 flanged plate

(*Note:* In practice any girder of the same total S.M. could be substituted)

Stanchion "h"	11(2b)	...	16′

Item	Section	Table	Size
"W"	11(2b)	. . .	20′ × 18.5′ × 16′ × 0.02 = 119
l	11(2a)	. . .	17′1″
Stanchion size	11(2)	12	$10\frac{3}{4}$″ × 0.593″ pipe
Upper deck beams, etc.			
"N"	10(2b)	. . .	2.5′ × 0.56′ × 7.5′ = 10.5
Beam and girder same as second deck			
Stanchion "h"	11(2b) and 10(2b)	. . .	9.5′
"W"	11(2b)	. . .	20′ × 18.5′ × 9.5′ × 0.02 = 70
"l"	11(2a)	. . .	7′4″
Stanchion size	11(2)	12	$8\frac{5}{8}$″ × 4.06″ pipe
Transverse bulkhead. Assume 30″ spacing of stiffeners			
Hold stiffeners "N"	12(4c)	. . .	2.5′ × 18.7′ × 0.35 = 16.4
(Bracketed) "l"	12(4c)	. . .	18′9″
Size	12(4c)	5	9″ × 3.45″ × 0.400″ channel
'Tween deck stiffener "N"	12(4c)	. . .	2.5′ × 4.5′ × 0.56 = 6.3
(Clipped) "l"	12(4c)	. . .	9′
Size	12(4c)	5	5″ × 3″ × $\frac{5}{16}$″ angle
Floor stiffener "N"	12(4c)	. . .	2.5 × 29.5′ × 0.7 = 62
"l"	12(4c)	. . .	3.5′
Size	12(4c)	5	6″ × $3\frac{1}{2}$″ × $\frac{5}{16}$″ angle
Plating. Write directly on Fig. 29	12(4a)	11	0.26″, 0.28″, 0.30″, 0.34″, (0.36″ + 0.04″)
Limber plate	12(4a)	11	(0.36 + 0.10)″
Floor plate	7(4c)	4	0.48″
Shell plating			
Freeboard to upper deck	. . .	. . .	30.25′ − 24′ = 6.25′
Table frame spacing	. . .	13	$27\frac{1}{2}$″
Increase for actual spacing	15(2a)	13	0.02″
Table draft	. . .	14	21′

Item	Section	Table	Size
Increase for actual draft	15(2a)	14	0.02″
Shell plating—bottom amidships	15(2a)	14	(0.60″ + 0.04″ = 0.64″)
Shell plating—bottom forward	15(2a)	13	1.2 T = 0.68″
Side shell, amidships	15(2a)	14	(0.58″ + 0.04″ = 0.62″)
Bottom and side at ends	15(3g)	13	0.45″
Sheer strake	15(2c, 3f)	14	50″ × (0.81 + 0.04″ = 0.85″) to 0.45″
Freeboard to bridge deck	. . .	. . .	37.75′ − 24′ = 13.75′
Bridge side plating	17(1a)	14	0.56″ + 0.04″ = 0.60″
Upper deck			
Gunwale angle	15(5)	. . .	6″ × 6″ × ⅝″
Area clear of bridge	16(2)	14	154 + (2 × 3.7) = 161 sq in
Area in way of bridge	16(2)	13	78″ + (2 × 2.0) = 82 sq in
Plating—minimum	16(4a)	15	0.39″
Plating abreast 20′ hatch	. . .	. . .	(161–7) ÷ [204″ + (2 × 3½″ = 0.73″ laps)]
Stringer plate—minimum		13	50″ to 30″ × 0.40″
Second Deck			
Area clear of bridge	16(2)	13	78″ + (2 × 2.0) = 82 sq in
Area in way of bridge	16(2)	13	72″ + (2 × 1.9) = 76 sq in
Thickness, minimum	16(4a)	15	0.34″
Thickness in way of 20′ hatch	. . .	. . .	82 ÷ [204″ + (laps = 2 × 2½″ = 0.36″)]
Stringer	16(3)	13	50″ to 36″ × 0.40″
Bridge Deck			
Area	16(2)	14	104 + (2 × 2.7) = 109 sq in
Stringer, minimum	16(3)	13	50″ × 0.40″
Plating, minimum	16(4a)	15	0.39″

CHAPTER VI

Problem 1. (a) 0.694. (b) 14,400 tons.
Problem 2. (a) 0.289. (b) 0.342. (c) 2740 tons.
Problem 3. $500 \times (65 + 72) \times 0.88 \times 52 \text{ lb} \div 2240 = 1400$ tons.
Problem 4.

	Solid floor, lb	Open floor, lb
Floor or bracket	1000	300
Frame	230	410
Reverse frame	190	390
Clip to keel	20	20
Stiffeners	110	...
Clip to margin	20	20
Strut	...	110
	1570	1250

Mean (50 percent of each).......1410 pounds
Times 2 for both sides...........2820 pounds
Divided by 2½-foot spacing.....1130 pounds per foot

Problem 5. (a) $10{,}000 \div 10.4 = 960$ tons.
(b) $(9000 \div 18) \times 10{,}000 \times 0.6 \times 1.15 \div 2240 = 1540$ tons.

CHAPTER VII

Problem 1. (a) 26.8 knots. (b) 7.11 feet per second.
Problem 2. 2.3 feet.
Problem 3. 1150 feet.
Problem 4. Acceleration, 143 pounds; static, 250 pounds; wind, 100 pounds; total, 493 pounds.

CHAPTER VIII

Problem 1. 10,440 tons, 211.6 feet *LCG* from *FP*.
Problem 2. $h = 8.0$ tons per foot. $x = 2.13$ tons per foot. $y = 3.20$ tons per foot.
Problem 3. (a) 201 square inches × feet. (b) 193 square inches × feet.
Problem 4. (a) 32.8 sagging. 36.2 hogging.

(b) sagging { keel 7.30* tension. / deck 7.61 compression.

hogging { keel 4.95 compression. / deck 8.00* tension.

Chapter IX

Problem 1. 6080/(60 × 33000) = 1/325.7.

Problem 2. The *ehp* of the destroyer is twice that of the freighter.

Problem 3. (a) 3.2 knots. (b) 16 knots. (c) 0.8.

Problem 4. $34690 \div \sqrt{11710 \times 415} = 15.74$.

Problem 5. B/draft = 2.25. Prismatic, 0.775. Displacement-length ratio, 164.

Correction to R_f = 1.019 × 15.74/15.4 = 1.0415.

$ehp = 11710 \times R/\Delta \times V \div 325.7$.

$V/\sqrt{L}$	0.60	0.65	0.70	0.75
V (knots)	12.22	13.24	14.26	15.28
Rr/Δ (Figs. 72 to 75)†	0.92	1.17	1.79	2.80
Rf/Δ (Fig. 69)†	2.50	2.89	3.32	3.78
Rf/Δ corrected	2.60	3.01	3.46	3.94
Total R/Δ	3.52	4.18	5.25	6.74
ehp	1550	2000	2700	2710

Chapter XI

Problem 1.

Tabular freeboard for 420-feet *LBP*	77.8 inches
Corrected for 0.76 block coefficient	82.5 inches
Depth correction	+6.9 inches
Effective length of superstructure (all 100 percent)	195 feet
Superstructure correction 33 percent of 42 inches.....	−13.8 inches
Total freeboard	75.6 inches or 6 feet 3⅝ inches

* Includes 18 percent increase for rivet holes.
† Taylor's "Speed and Power of Ships."

Allowable draft (30 feet $3\frac{5}{8}$ inches) − (6 feet $3\frac{5}{8}$ inches) = 24 feet

Problem 2. (a) Yes. (b) No. (c) Yes, either the forepeak bulkhead, the after peak bulkhead, or the bulkhead between the deep tanks and the machinery space. (d) One-compartment.

Problem 3. Stability flooded

Lost Buoyancy	Volume, cubic feet	*VCG*, feet
55 × 54 × (20 − 3.5) at 80 percent	39,200	11.75
[35 × 54 × (20 − 3.5) − 15000] at 50 percent	8,090	11.75
Net lost buoyancy	47,290	11.75
÷ 35 =	1351 tons	

Lost waterplane: 90 feet × 54 feet × 95 percent = 4610 square feet.

Lost tons per inch: 4610 ÷ 420 = 11.0 tons

Remaining tons per inch: 45.1 − 11.0 = 34.1.

Sinkage: 1351 ÷ (34.1 × 12) = 3.30 feet.

Mean height of new layer: 20 feet + ½(3.30) = 21.65.

Rise of lost buoyancy: (21.65 − 11.75) = 9.90 feet.

Rise of *VCB:* 9.90 feet × 1351/9570 = 1.40 feet.

Loss of inertia: 90 feet × $(54)^3$ × 95 percent ÷ 12 = 1,120,000 feet^4.

Loss of *BM:* i/V = 1,120,000 ÷ (35 × 9570) = 3.24.

Net loss of *GM:* 3.24 feet − 1.40 feet = 1.84 feet.

(a) Flooded *GM:* 3.5 feet − 1.84 feet = 1.66 feet.

Heeling moment 15,000 × 0.50 × 13.5 ÷ 35 = 2893 foot-tons.
Tangent $\theta = wd/\Delta GM$ = 2893 ÷ (9570 × 1.66) = 0.182.
(b) Angle of heel θ = about 10.3 degrees.

CHAPTER XII

Problem 1. (a) 388 feet. (b) 640 tons.
Problem 2. 7.47 tons per square foot.

TABLE 4(A)

Horsepower Calculation for Problem Ship

Displacement = 14,740 tons *WS* = 40,800 sq ft B/Dr = 2.19
Prismatic coefficient, 0.728 Displacement-length ratio, 156
Length correction to R_f (Fig. 39) for 455 feet = 0.995

		Rr, lb per ton							
$V/\sqrt{L}$	*V*, kts	2.25	3.75	2.19	ehp_r	$\frac{ehp_f}{1000\text{ sq. ft.}}$	ehp_f	ehp_{BH}	shp_{sea}
0.60	12.87	0.85	1.09	0.84	490	38.5	1560	2050	4000
0.65	13.94	1.08	1.50	1.06	670	48.4	1970	2640	5150
0.70	15.01	1.48	2.07	1.46	990	60.0	2440	3430	6700
0.75	16.09	2.06	2.90	2.03	1480	72.7	2950	4430	8650
0.80	17.16	3.15	3.90	3.12	2420	87.0	3530	5950	11600

$$ehp_r = Rr/\text{ton} \times \Delta \times V \div 325.7 = Rr/\text{ton} \times V \times 45.26$$

$$ehp_f = \frac{ehp_f}{1000\text{ sq ft}} \times \text{length correction} \times \frac{WS}{1000} = \frac{ehp_f}{1000\text{ sq ft}} \times 40.6$$

$$shp = (ehp_{BH} \times 1.05 \div 0.62) \times 1.15 = ehp_{BH} \times 1.95$$

1.05 is allowance for appendages (rudder and bilge keels).
0.62 is assumed propulsive coefficient.
1.15 is allowance for sea power.

TABLE 5(A)
Ordinates for Section Area Curve

Sta.	S.A., from Fig. 65	S.A., faired	S.M.		S.A., for correct prismatic	Half area, sq ft
0	0	0	× 1 =	0	0	0
1	0.178	0.178	× 4 =	0.712	0.178	139
2	0.413	0.410	× 2 =	0.820	0.409	319
3	0.640	0.637	× 4 =	2.548	0.635	495
4	0.810	0.810	× 2 =	1.620	0.808	629
5	0.923	0.925	× 4 =	3.700	0.923	719
6	0.980	0.980	× 2 =	1.960	0.979	763
7	1.000	1.000	× 4 =	4.000	1.000	779
8	1.000	1.000	× 2 =	2.000	1.000	779
9	1.000	1.000	× 4 =	4.000	1.000	779
10	1.000	1.000	× 2 =	2.000	1.000	779
11	1.000	1.000	× 4 =	4.000	1.000	779
12	0.998	0.992	× 2 =	1.984	0.991	772
13	0.975	0.970	× 4 =	3.880	0.968	754
14	0.928	0.925	× 2 =	1.850	0.923	719
15	0.845	0.847	× 4 =	3.388	0.844	657
16	0.730	0.730	× 2 =	1.460	0.728	567
17	0.580	0.574	× 4 =	2.296	0.572	446
18	0.385	0.385	× 2 =	0.770	0.384	299
19	0.190	0.190	× 4 =	0.760	0.190	148
20	0	0	× 1 =	0	0	0
			Sum =	43.748	43.684	
Prismatic coefficient = sum ÷ 60 =				0.729	0.728	

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5th Sept